POWER PLANT ENGINEERING

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Uneeda Bakers Start Up New Buffalo Plant

NATIONAL BISCUIT COMPANY HAS CONSTRUCTED A MODERN INDUSTRIAL POWER PLANT TO MEET ALL REQUIREMENTS OF A LARGE BAKERY

FIRST IMPRESSIONS may not always be lasting; however, the new power plant of the National Biscuit Co. at Buffalo gives the visitor an impression of good design and neatness that will not soon be forgotten.

Two Skinner Universal Uniflow engine-driven generator units have been installed to take care of the present power and a part of the low pressure heating steam requirements. Space has been provided for a third engine unit which will be installed at a future date. Each engine is 20 by 24 in. and is rated at full load and 200 r.p.m. at 320 i.hp. Operating conditions are specified as 150 lb. ga. pressure and 100 deg. F. superheat at the throttle. Under these conditions, the guarantee of economy is as follows:

Load, i. hp.	¼	½	¾	Full	1¼
Lb. steam per i. hp	18.7	17.6	17.3	17.4	18.3
Mechanical Eff	80.7	89.5	91.5	93.0	94.5

Direct connected to each engine is a 3-phase, 60-cycle, 225-v. General Electric Co. generator, rated at 200 kw. Excitation may be obtained from either a motor generator set with direct connected exciter, or a turbine driven generator exciter. Both exciters are rated at 125 v. and 35 kw. The economy guarantee of the turbine driven unit is as follows:

Output of generator kw	46.6	35.0	17.5
Lb. steam per kw.-hr	55.0	59.5	78.0

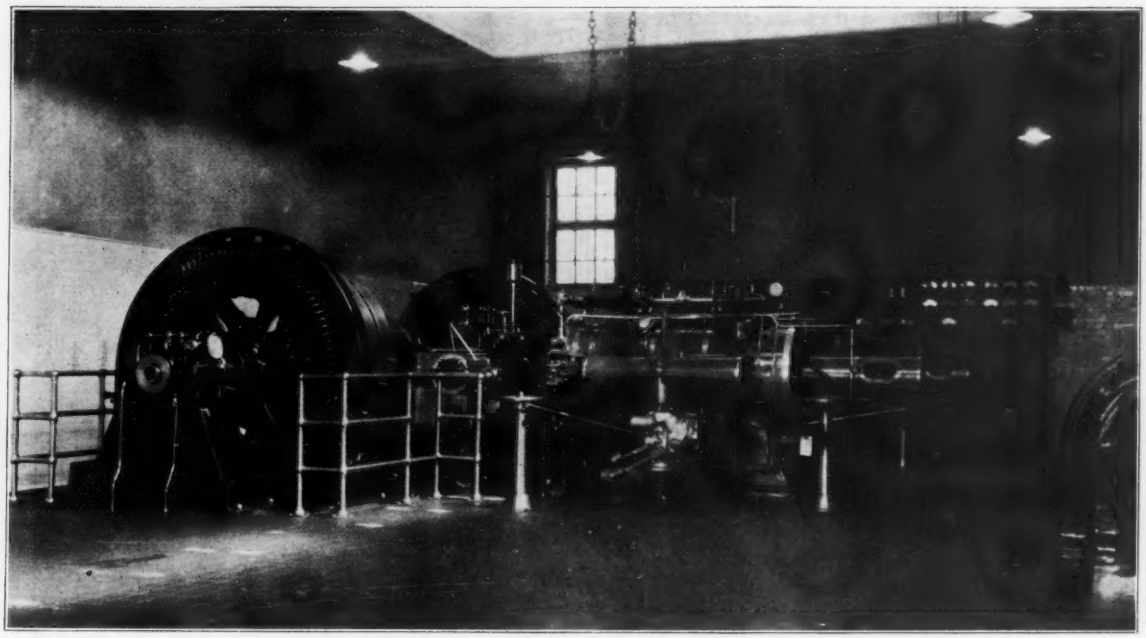


FIG. 1. TWO ENGINE-DRIVEN GENERATORS HAVE BEEN INSTALLED

This unit will be reserved for starting up the plant and as a spare in case of failure of the motor-generator unit.

Each engine is equipped with a 7-in. Strong, Carlisle and Hammond stop valve in addition to the regular throttle valve. The stop valves are conveniently located above the floor and each is operated either by means of an overspeed trip device mounted on the generator shaft or by means of push buttons, one of which is located at the switchboard and one along the east wall to the south. In this way either engine may be cutoff from high pressure steam in a moment's notice.

The main switchboard is located along the north wall just back of engine unit No. 1. It serves the two

main generators and the distribution system. An auxiliary switchboard provides switching facilities for the motor of the motor-driven exciter, the synchronous motor which drives the ammonia compressor and the small motor-driven exciter which is for use with this compressor. This small exciter set was manufactured by the General Electric Co. Its motor is rated at 220 v., 48 amp., 1750 r.p.m. and 20 hp. and the compound-wound generator is rated at 125 v. and 80 amp.

One Frick 55-T. ammonia compressor has been installed. This machine is direct connected to a General Electric synchronous motor, rated at 220 v., 40 deg. C., 150 r.p.m. and 125 hp. When operating this motor it is possible to hold the plant power factor at unity; when it is not in operation the power factor averages about 93. Power factor correction is also obtained by the 150-kv.a. synchronous motor which drives the 35-kw. exciter; the high rating of the motor permitting a wide variation in the reactive load.

Lubrication of all main units, including the feed pumps, is provided for by means of a central system of

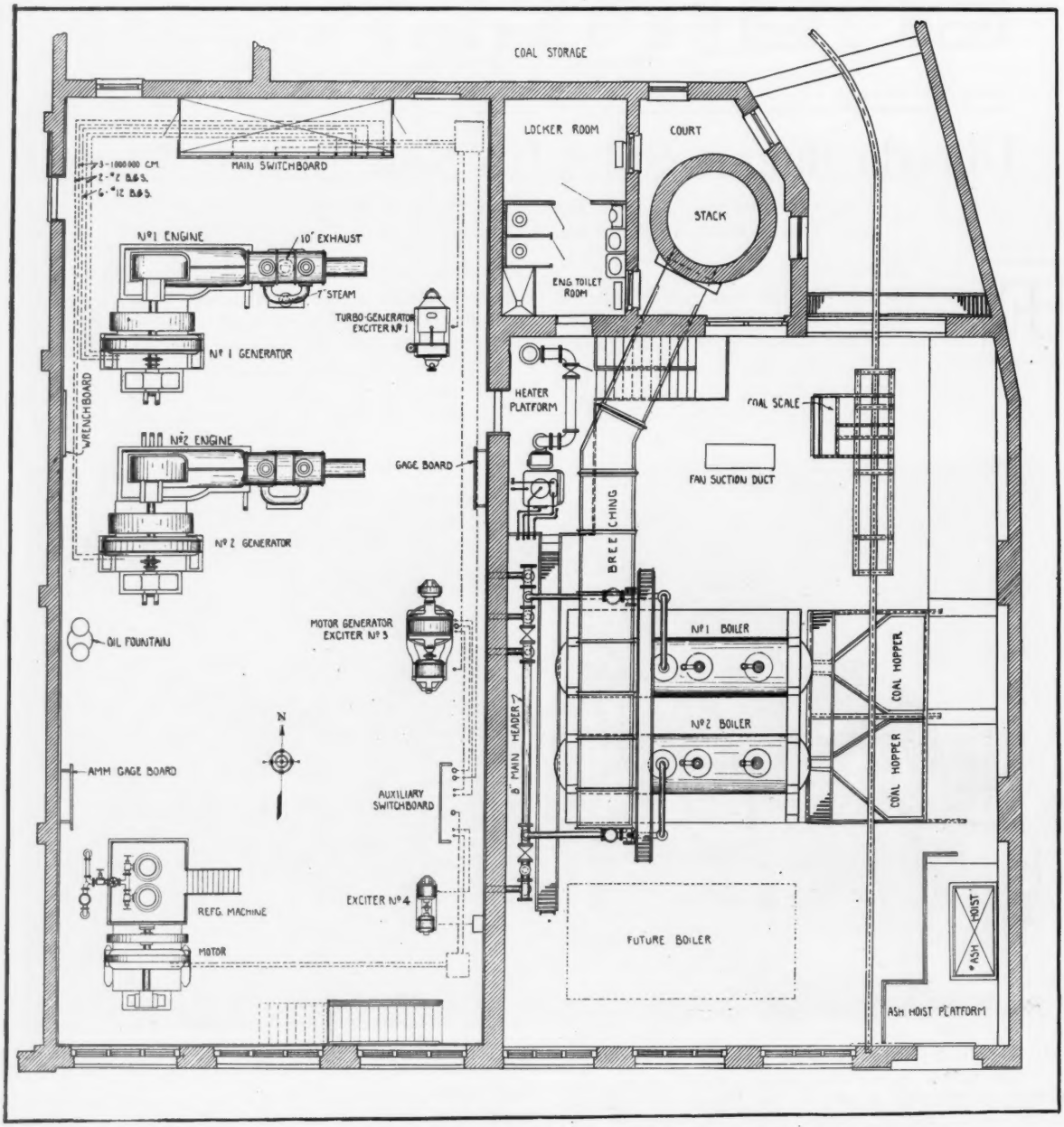


FIG. 2. FUTURE ADDITIONS OF ONE ENGINE UNIT AND ONE BOILER HAVE BEEN PROVIDED FOR

Bowser make. Two tanks, one for cylinder and one for bearing oil, are located overhead in the boiler room. The oil flows by gravity to the point of application. That oil which can be used again is filtered and pumped back to the overhead tank. This scheme leaves the bearing lubrication system provided with each engine as spare units, which will be used only in case of emergency. An oil fountain is located near No. 2 generator and two float gages, mounted on the engine wall near

the auxiliary switchboard, indicate the height of the oil in the two overhead tanks.

To bring as much as possible of the plant control under the management of the engine room operator, the valve stems for the main high pressure boiler room valves have been planned so that they extend through the wall into the engine room. The hand wheels are then mounted in the engine room at a convenient height so they can be easily reached from the floor. The valves brought in this manner under the engineer's control are: one main steam header sectionalizing valve, two engine lead valves, two boiler lead valves and two auxiliary header valves. The four valves operated from

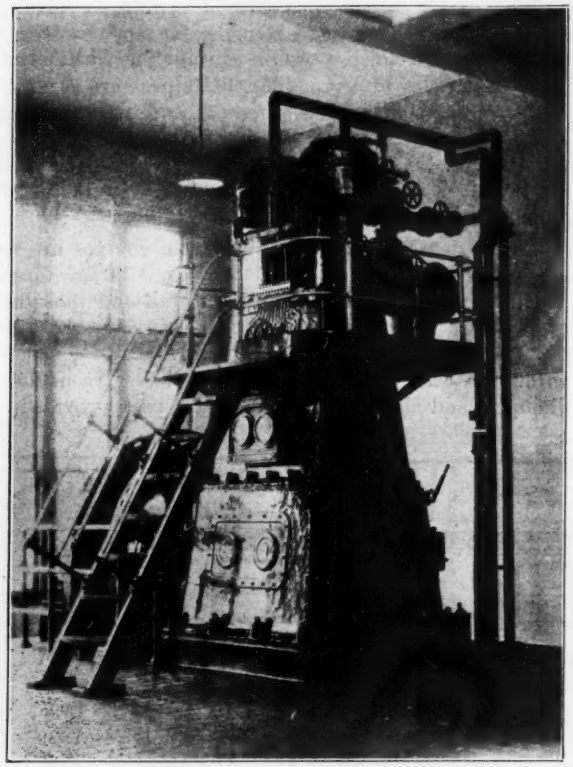


FIG. 3. POWER FACTOR CORRECTION IS PROVIDED BY SYNCHRONOUS MOTOR-DRIVEN AMMONIA COMPRESSOR

the engine floor room stands are: two engine throttle valves with bypasses, and two engine exhaust valves.

Two Republic CO_2 recorders are conveniently located in the engine room near the door leading to the boiler room. On the opposite side of this door is a four-paneled slate board on which are mounted one Republic flow meter, a back pressure indicating gage, a steam pressure indicating gage, two stack temperature recorders, an exhaust steam pressure recorder, and recorders which show the feed water temperature, steam pressure and the total temperature of the steam.

An alarm is provided in the engine room by means of which any elevator operator in the building may signal to the engineer.

An ammonia gage board is located convenient to the compressor. The visitor will find hanging on the engine room wall five neatly framed licenses to prove that this plant is well watched over.

Daylight is provided for the engine room by windows in the north and south walls; in addition there is a monitor roof section which covers a large part of the room. This affords additional daylight and provides an excellent means of ventilation. The electric lighting fixtures are in keeping with the general appearance of the room and are ample in number for good illumination when required.

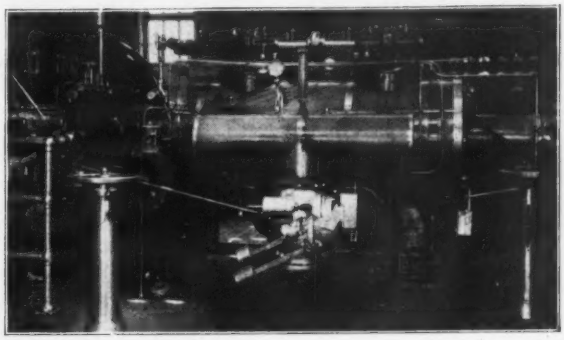


FIG. 4. EACH ENGINE IS EQUIPPED WITH AN AUTOMATIC STOP VALVE

Glazed white tile extends up from the floor about 8 ft. Above this the walls are finished off in a buff, all of which makes a very effective appearance in contrast with the red tile flooring. Three exits are provided from the engine room, one leading to the bakery, one leading to the boiler room and one leading to the basement. It would be hard to conceive of an emergency during which a man could not readily find a safe exit from this room.

Entrance to the boiler room leads to a platform which is above the boiler room floor. It is at the level of the engine room floor and at about the height of the boiler drum center line. On this platform is located

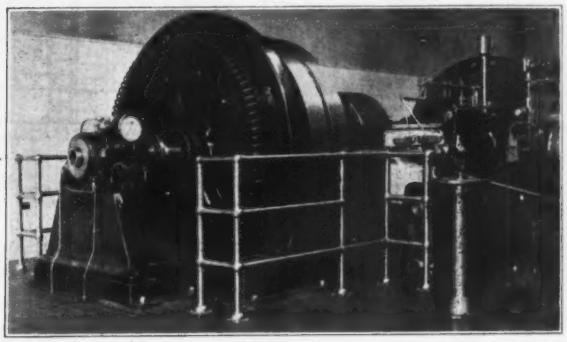


FIG. 5. SUBSTANTIAL GUARD RAILING HAS BEEN PLACED AROUND MAIN GENERATING UNITS

an 800-hp. Worthington open type feed water heater. This heater is of the induced type, the steam connection to which is made in the vertical run just below the back pressure valve.

Steps then lead down to the boiler room floor where two 206-hp. Babcock & Wilcox water-tube boilers have been installed. Space has been provided for the addition of a future unit. These boilers operate under 160 lb.

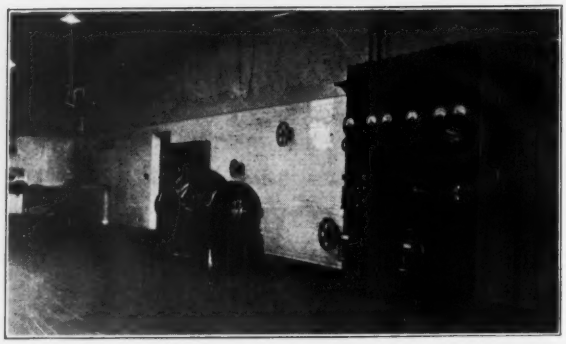


FIG. 6. STEMS FOR BOILER ROOM STEAM LINE VALVES ARE EXTENDED THROUGH THE WALL TO THE ENGINE ROOM SIDE

gage pressure and deliver steam superheated to 100 deg. F. by means of Foster superheaters. The ratio of superheater surface to boiler heating surface is 21 per cent.

These boilers are set with the front header about 10 ft. above the operating floor and the baffles are arranged for three gas passages. Combustion Engineering Corp. type E class 6 stokers have been installed. These stokers are 6 ft. 6 in. long by 6 ft. 2 in. wide, which provides 36.7 sq. ft. of effective grate surface for each boiler. The ratio of boiler water heating surface to grate surface is 56.2 to 1 and the volume of the combustion chamber per square foot of projected grate area is 7 cu. ft.

Stoker performance guarantees were based on an ultimate coal analysis, as follows:

Moisture	3.5 per cent
Volatile matter	35.0 per cent
Sulphur	2.0 per cent
Ash (dry basis)	8.0 per cent
B.t.u. (dry basis)	13,000 per lb.

An analysis of the coal which is actually being burned is:

Moisture	3.38 per cent
Volatile matter	38.70 per cent
Fixed carbon	49.30 per cent
Ash (dry basis)	12.00 per cent
B.t.u. as fired	12,613 per lb.

Each boiler is equipped with a Strong, Carlisle & Hammond triple-acting non-return valve, an "S-C" feed water regulator which works in conjunction with Mason pump governors under a differential pressure of from 15 to 20 lb., Diamond soot blowers and an indicating Precision draft gage which shows normally 1 in. of water under the fire, 0.18 in. draft at the damper and a minus 0.08 in. of water over the fire. Soot is blown from the tubes twice each 24 hr.

Two 5 by 7-in. vertical Troy steam engines are set so as to drive one 9000 cu. ft. per min. Clarage forced draft fan. One engine operates the fan and the other is held as a reserve unit by disconnecting the shafts at the coupling. Air from the fan is carried to the stokers through a concrete duct beneath the boiler room floor. Dampers and clean out openings have been provided at

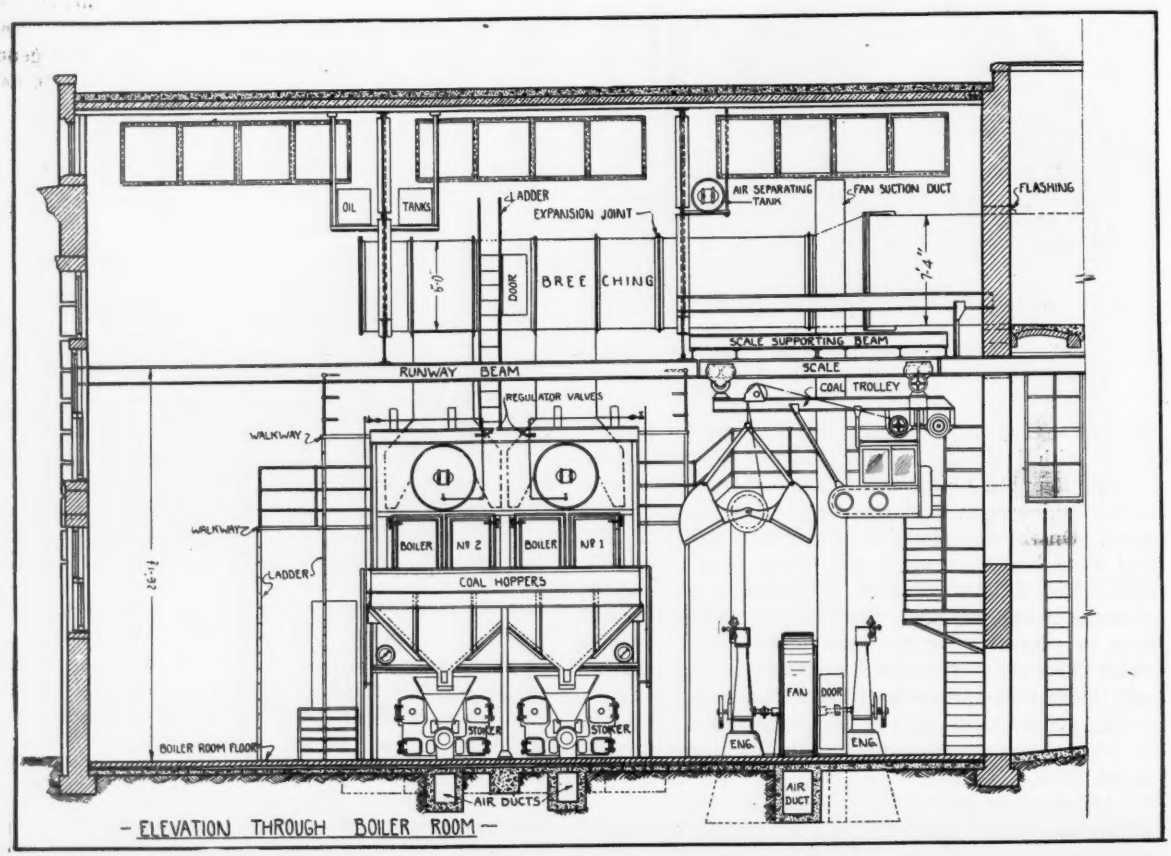


FIG. 7. COAL IS WEIGHED BY MEANS OF A MONO-RAIL TRACK SCALE

suitable points in the duct. Air for the fan is taken by means of a duct from near the boiler room ceiling. In this way a slight economy is effected.

Automatic regulation of furnace conditions is obtained by means of a Mason regulator attached to the fan engine. With a decrease in steam pressure this regulator causes the fan engine to speed up, in this way increasing the amount of air flow and also the pressure under the stoker and in the combustion space. The balanced draft apparatus, manufactured by The Engineer Co., then acts due to the increased pressure in the furnace and opens the damper and speeds up the stoker.

Both boilers discharge their waste gases into a common sheet metal breeching, which is heavily insulated on the outside. A radial brick stack 5 ft. in diameter by 150 ft. high is of sufficient capacity for three boilers. The breeching opening to the stack is 4 ft. wide by 7 ft. 4 in. high.

There is no basement under the boiler room; the fireman rakes the ashes out on the floor from beneath the stoker and loads them into cans which are moved but a short distance onto an electric elevator. The lift of this elevator is about 10 ft. This brings the ash cans to the height of a platform a few feet above the street level. From this platform the ash is dumped into a motor truck and hauled away.

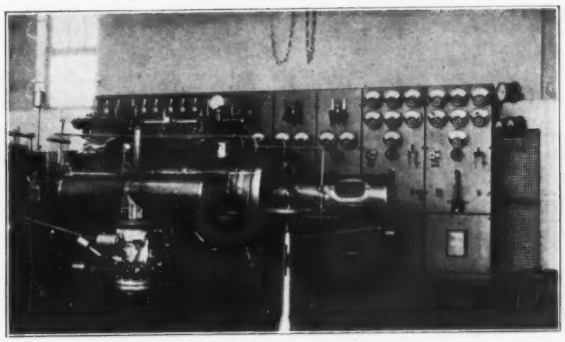


FIG. 9. VIEW SHOWING RELATIVE POSITION OF THE SWITCHBOARD AND THE FIRST ENGINE UNIT

Coal is delivered to the plant in railroad cars and dumped directly into an enclosed coal storage room right alongside of the boiler room. This scheme is particularly advantageous because the track grade is about 20 ft. above the boiler room and coal storage room floors which are on the same level. Two grades of coal are purchased, one for use in the boilers and one for use in gas producers, the gas from which is used for heating the baking ovens. As the coal is loaded into

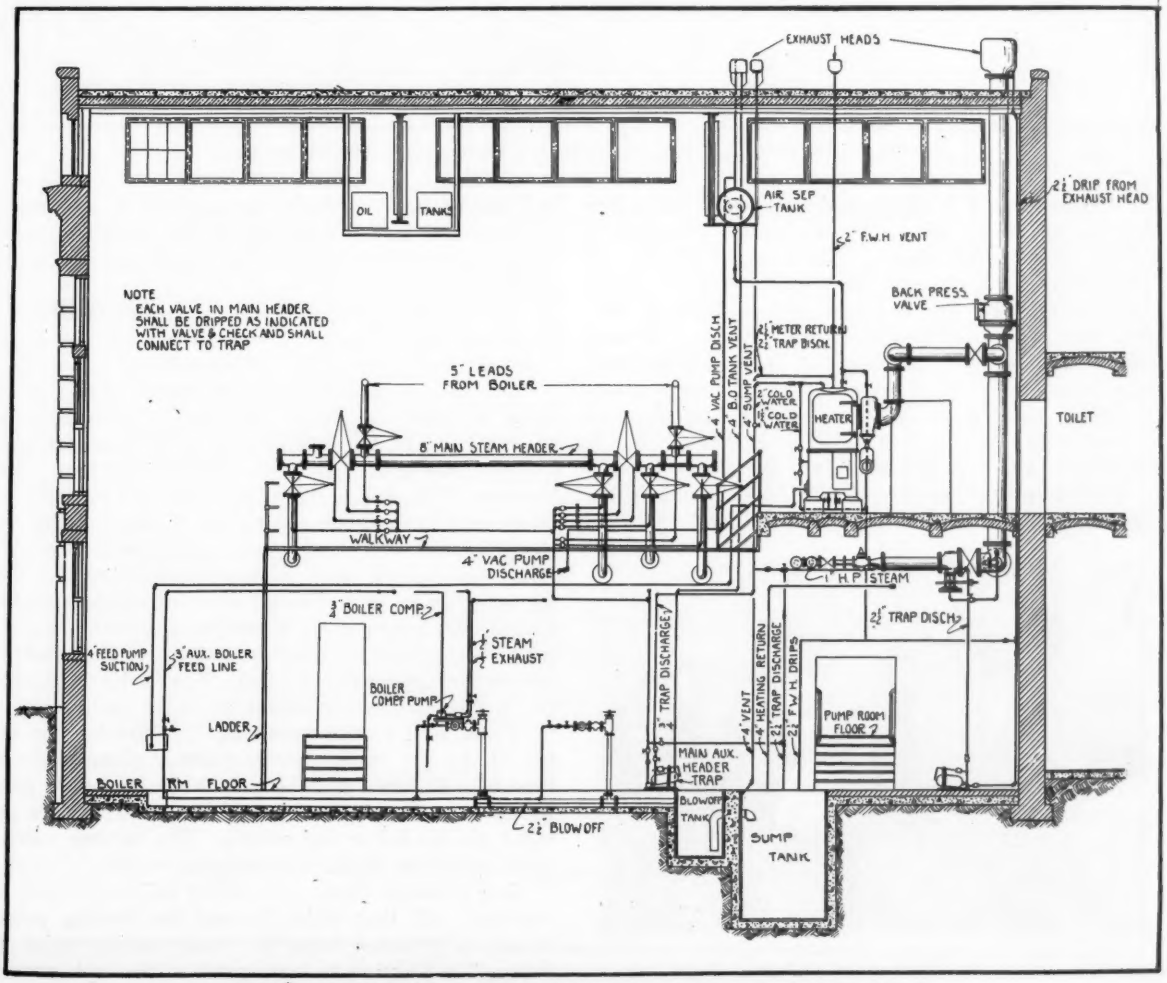


FIG. 8. ELEVATION OF MAIN STEAM PIPING AND FEED WATER HEATER

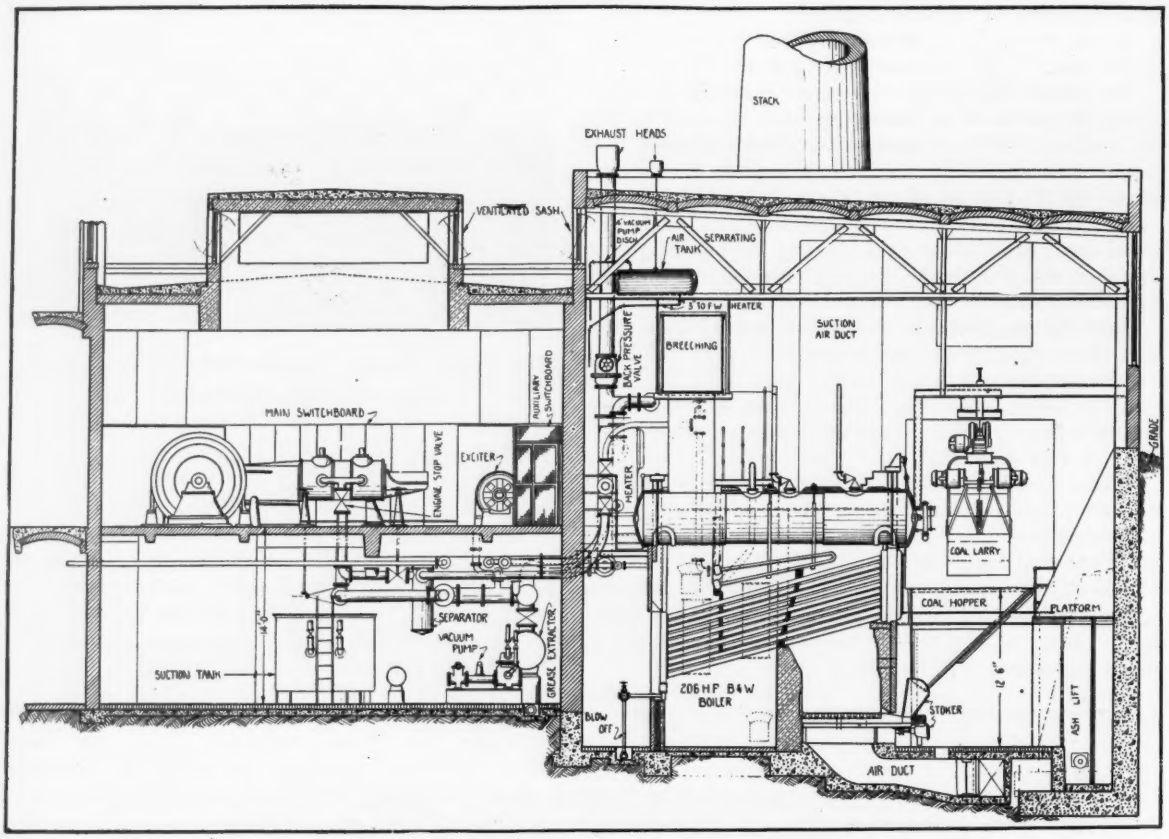


FIG. 10. SECTIONAL ELEVATION THROUGH ENGINE AND BOILER ROOMS

the storage room, it is distributed or taken to the stoker hoppers by means of a 1½-yd. bucket attached to a Shepard mono-rail crane. The storage capacity of boiler coal is 800 T. and each stoker hopper has a capacity of 10 T. The mono-rail system makes a complete loop in the coal storage room starting at the coal dump chute and passing through an opening in the partition wall into the boiler room. Just as the crane enters the boiler room one section of the track is fitted up as a scale beam; in this manner, each bucket of coal is weighed before it is unloaded into the stoker hopper.

An interesting feature of this boiler plant is the complete and substantial system of ladders, walkways and handrails. Ventilation is provided for by means of ventilated sash. Windows in the north and south walls, together with the monitor sash, provide ample daylight.

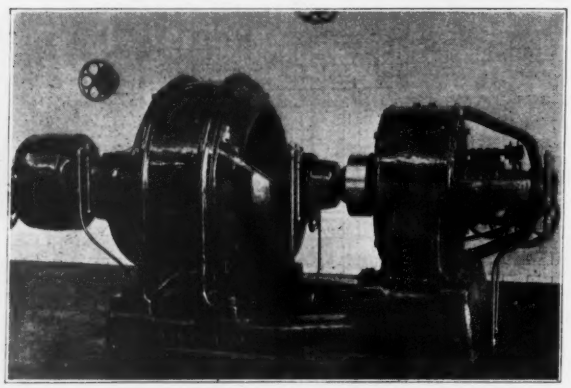


FIG. 11. MOTOR GENERATOR SET PROVIDED WITH DIRECT CONNECTED EXCITER

On the same elevation with the heater is located a toilet and locker room. A separate steel locker is provided for each employe of the engine and boiler room.

Two doorways, one under the heater platform and one near boiler unit No. 3, lead into the basement under the engine room, the floor level of which is on a slightly higher grade than that of the boiler room floor. This basement is known as the pump room and contains two boiler feed pumps, one electric-driven house pump, one steam-driven house pump, one vacuum pump, a sprinkler filling pump, and three Westinghouse locomotive type air compressors, two of which are used on the sprinkler system, and one operates in conjunction with the Johnson service system for temperature control of the heating system throughout the building. Ventilation is provided for in this basement by means of an Ilg wall fan.

Boiler feed pumps were made by Worthington and are of the pot valve, outside packed, plunger, duplex type, size 5¼ by 7½ by 4½ by 10 in. and each pump is rated at 132 gal. per min. Mason governors are provided for the boiler feed pumps. The vacuum pump is an 8 by 12 by 12-in. Worthington.

Low-pressure steam for heating the feed water, the building, and that which is used for heating process water, is provided from the main engine units, the reduced in pressure to correspond to the back pressure carried and cut into the low-pressure steam main.

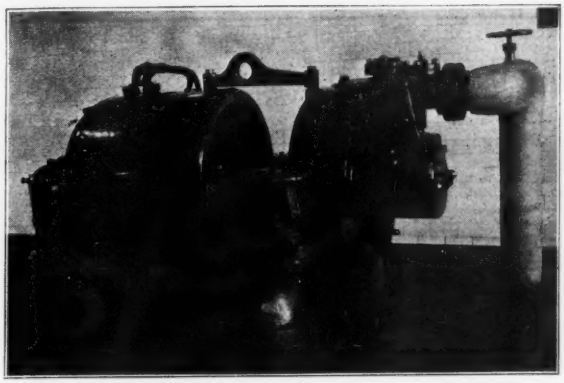


FIG. 12. 35-KW. STEAM-DRIVEN EXCITER SET

steam-driven exciter, the stoker fan engine and the steam-driven pumps and air compressors. In winter about 40 boiler horsepower of high pressure steam will be required in addition to the foregoing. This will be

Drips from the high-pressure header, auxiliary header and the low pressure steam mains are handled through Crane Tilt traps which discharge to the feed water heater. All return water from the heating system is returned to the heater by means of a low pressure lifting trap.

City water, for boiler feed make up and house service is first brought into storage tank and from there it is pumped to a gravity tank on top of the building by means of the house or service pumps. Make-up water is fed directly to the heater. Dearborn feed water treatment is fed into the hot water main between the boilers

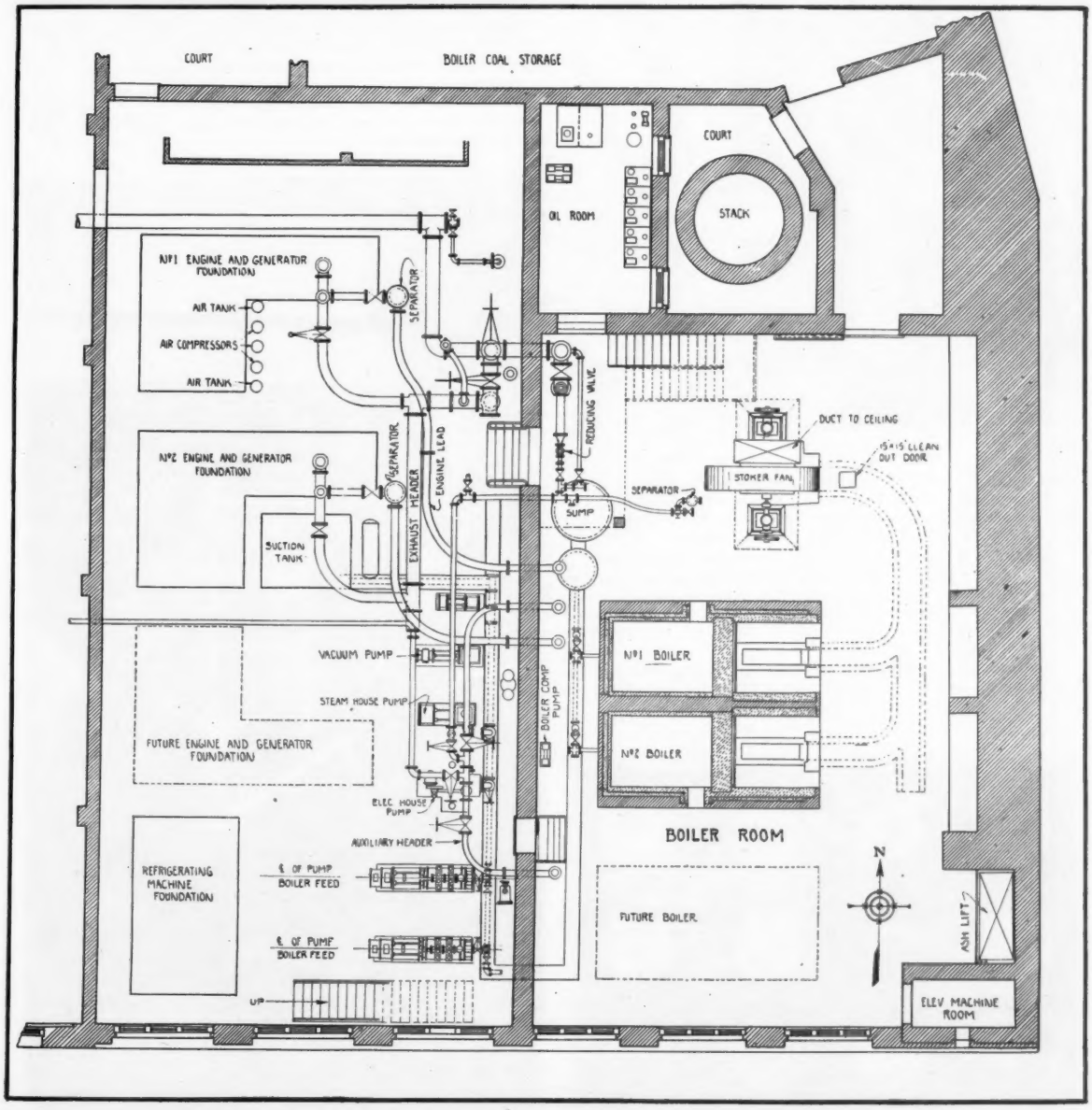


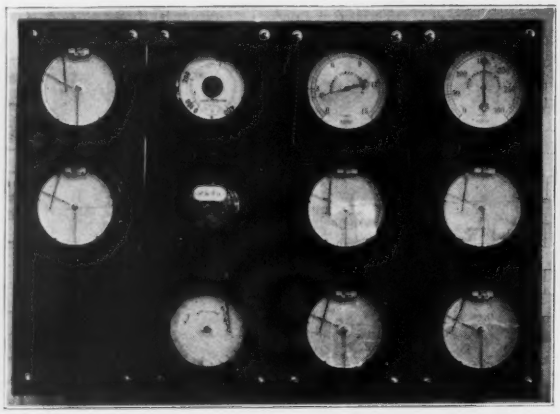
FIG. 13. PIPING PLAN IN BOILER ROOM AND BASEMENT OF ENGINE ROOM

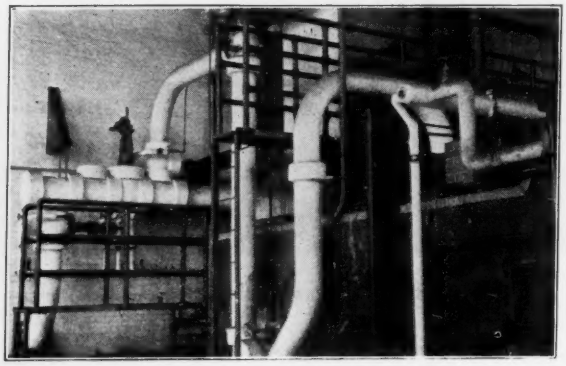
Fig. 14. Some Details of the Boiler Room Layout

Coal is delivered to the stoker hoppers by the mono-rail coal larry as shown in the picture above. The mono-rail system is arranged to distribute the coal in the storage room and to reclaim it is required for power purposes.

One section of the track, just inside the boiler room, is arranged as a scale beam so that each bucket of coal may be weighed before it is dumped into the stoker hopper. In this way it is possible to keep an accurate record of the coal used for each boiler over a given period of time. Each stoker hopper has a capacity of 10 T.

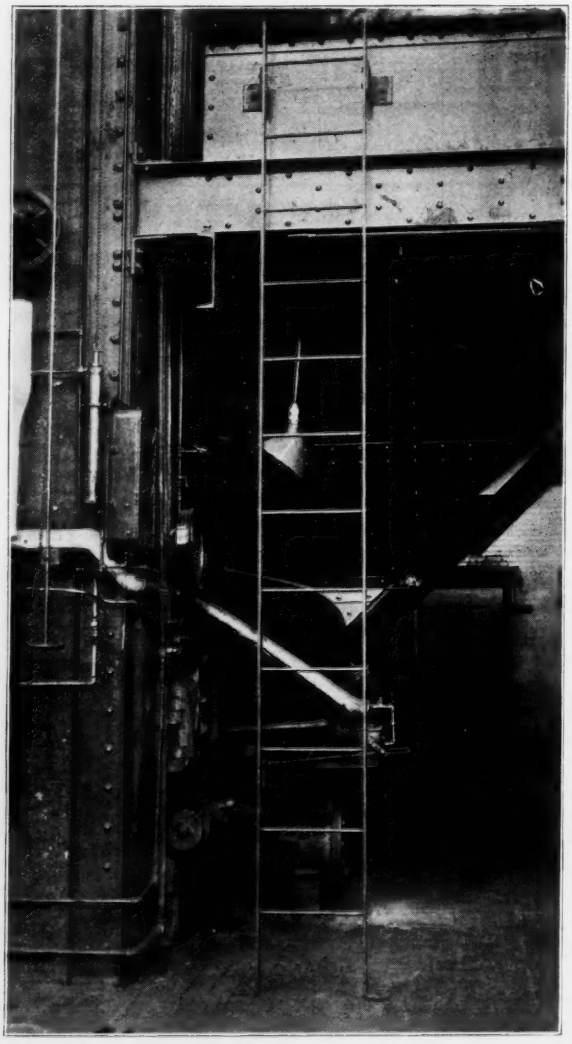
Instruments as shown mounted in the view below are for checking up and maintaining records of boiler operation.





Ladders and runways as shown in the view above are substantial in their design and are provided so that they are a direct advantage in operation and maintenance.

Draft and steam pressure indicating gages, as shown in the picture below, are mounted on the boiler frame at a convenient height so that they are under the close observance of the operator at all times.



and the Ross filter which is located just after the feed pumps.

There is always a certain amount of waste hot water which contains grease, chocolate, etc. The heat in this water is reclaimed by passing it through a tank in which are located coils. The boiler feed make-up water is passed through these coils before going to the heater. All water used for dough mixing is passed through a Loomis-Manning filter, the beds of which are made of bone char and sand.

Located a short distance from the engine room and in the basement of the bakery is the ammonia condenser, which consists of seven banks of coils, two sections high. Cooling water passing from the condenser may be diverted to a storage tank below the floor, from which it may be pumped for general purposes throughout the building; or it may be recirculated after it has been run through a cooling tower located on the roof. Two pumps have been provided to circulate the cooling water and two brine pumps have been installed, one of which supplies cold brine to the plant and the other circulates the brine through the expansion tank.

This power plant is operated in three 8-hr. shifts, one engineer and one fireman are on duty during each shift.

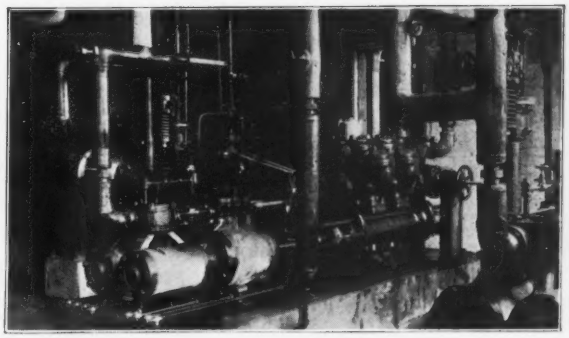


FIG. 16. BOILER FEED PUMPS ARE LOCATED IN THE ENGINE

During the day shift, one oiler is employed. A fourth or relief crew, consists consists of one engineer and one fireman, who during their relief period of one week report at the plant on three days of that week and act in the capacity of repairmen or maintenance men.

This plant can well stand as a model of the industrial type. Details prerequisite to convenience and efficiency have not been overlooked, as is evidenced by the extreme care exercised by the designers. A great deal of attention has been given to the matter of safety, as in the

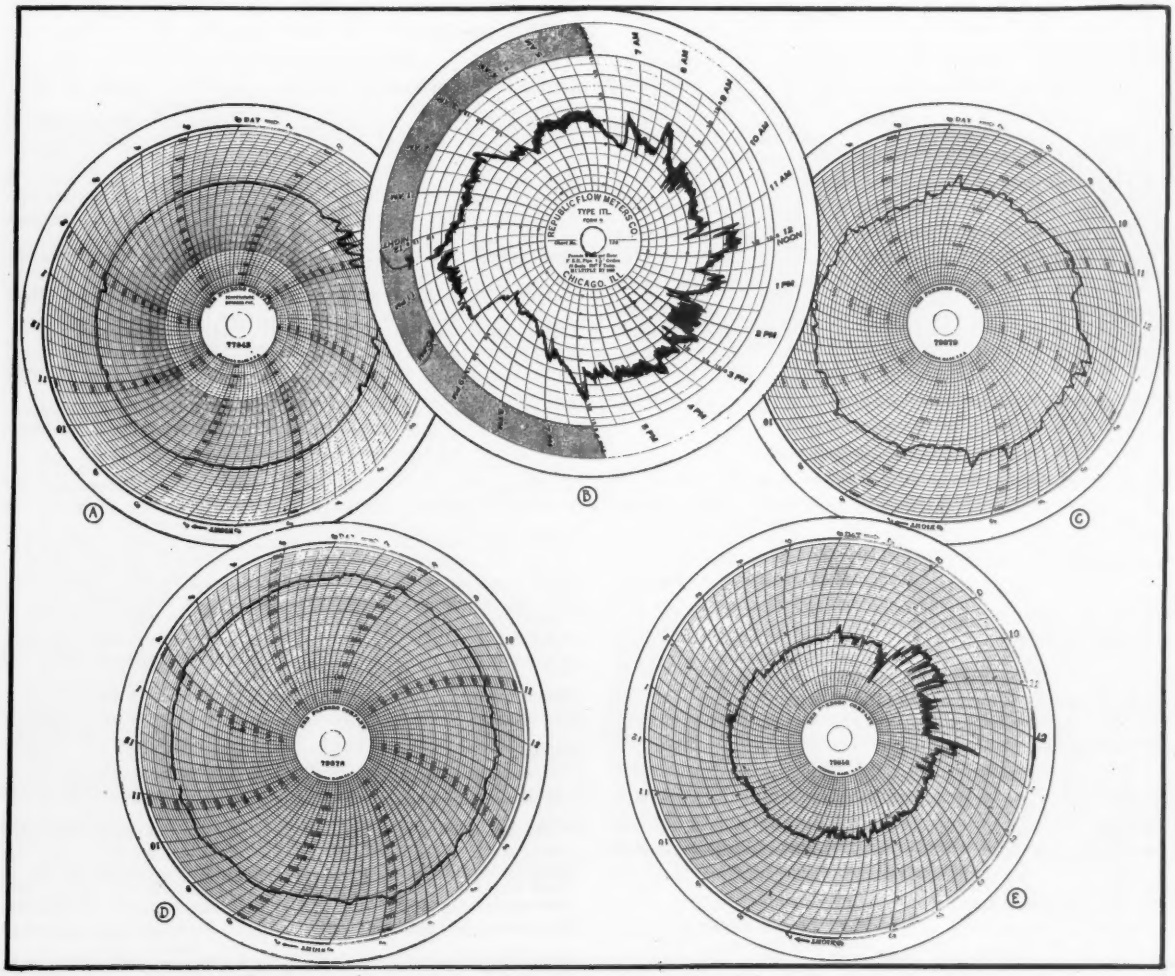


FIG. 15. RECORDERS PLAY AN IMPORTANT PART IN THE ECONOMY OF OPERATION

(A) Feed water temperature averages about 217 deg. F. (B) A flow meter records the amount of water fed to the boilers. (C) Record of daily stack temperatures. (D) Total steam temperature averages about 485 deg. F. (E) Record of the back pressure against which the engines operate.

provision of remote control for the engine stop valves, the complete system of substantial and prominent runways, the excellent means of assuring proper lubrication for the main units, and in the provision of effective ventilation, the effective use of day light and in the provision for proper conditions of sanitation.

Provision for taking full advantage of day light is too often overlooked by designers of engine and boiler rooms. The problem should be looked upon, not only as regards safety to individuals against personal injury, but as a provision which greatly promotes personal efficiency through making surrounding conditions more pleasant.

Another point well worth consideration is the spirit of pride in this plant which exists among the operators. They feel that everything has been laid out and provided for efficient operation and their convenience, and they have set out to show that they know how to make the best use of every piece of equipment within their charge. They are co-operating with all those who are concerned with the power plant, and the results are bound to be satisfactory to the management, no matter what angle they may be viewed from.

This plant was designed by D. M. McKelvey, consulting engineer, and is operated under the direct care of C. A. Myers, plant engineer.

MECHANICAL EQUIPMENT AT NATIONAL BISCUIT COMPANY'S BUFFALO PLANT

Boilers

Manufacturer	The Babcock & Wilcox Co.
Number	2, each 2065 sq. ft. ht. surface
Boiler working pressure	160 lb. per sq. in. gage
Ratio of superheater surface to boiler heating surface	21 per cent

Forced Draft Fans

Manufacturer	Clarage Fan Co.
Number	1
Type	"R" ¾ Hsd. No. 160
Speed	350 r.p.m.
Capacity, at 5 in. static pressure	9000 cu. ft. per min.
Drive	2—5" by 7" vertical Troy engines—1 standby

Stokers

Manufacturer	Combustion Engineering Corporation
Type	E, Class 6
Width of grate	6'—2" 36.7 sq. ft. effective
Length of grate	6'—6" grate surface
Ratio of water-heating to grate surface	56.2 to 1
Ultimate analysis of coal upon which stoker performance is based:	
Moisture	3.5 per cent Volatile 35%
Ash (dry basis)	8 per cent Sulphur 2%
Heat value (dry basis)	13,000 B.t.u. per lb.

Boiler Settings

Volume of combustion chamber per sq. ft. of projected grate area	7 cu. ft.
Arches	None

Engine-driven Generator Sets

Manufacturer, Engine	Skinner Engine Co., Universal Unaflow. 20x24", 200 r.p.m., 320 i.hp. rated full load
Manufacturer, Generator	General Electric Co., 3 ph. 60-cycle, 225 v.
Rating	200 kw.
Number of units	2
Operating conditions	150 lb. gage, 100° F. superheat

Guaranteed Performance of Units

Output of engine generator per i.hp	¼	½	¾	1	1¼	
Lb. of steam per i.hp	Load	18.7	17.6	17.3	17.4	18.3
Mechanical efficiency		80.7	89.5	91.5	93	94.5

Turbo-Generator Exciter

Manufacturer	General Electric Co.
Number	1
Capacity	35 kw.
Generator	125-v., direct current for excitation
Condenser	None
Output of Turbo:	
Generator Kw	46.6 Lb. of steam 55
	35.0 Per kw.-hr. 59.5
	17.5 78
M.G. set with direct connected exciter	General Electric Co. 125 v.
Rating	35 kw.

Feed Water Heater

Manufacturer	Worthington Pump & Machinery Corp.
Type	800 hp. Stilwell open type
Number	1

Boiler Feed Pumps

Manufacturer	Worthington Pump & Machinery Corp.
Number	2
Type	Pot valve, outside packed plunger type
Capacity, gal. per min	132

Miscellaneous Equipment

Ammonia Compressor	Frick Co., Inc., (5 t. capacity)
Coal handling	Shepard Electric Crane & Hoist Co. (1½ yd. bucket)
Coal hoppers	August Feine & Sons Co.
Superheaters	Foster Superheater Co. (sold by Power Specialty Co.
Soot blowers	Diamond Power Specialty Corp.
Automatic non-return valves	Strong, Carlisle & Hammond Co.
Steam separator	Direct Separator Co.
Throttle valve	Nelson Valve Co.
Pressure reducing valves	Boylston Steam Specialty Co.
Safety valves	(Consolidated) Manning, Maxwell & Moore Co.
Gages, recording, etc	The Foxboro Co.
Thermometers	Taylor Instrument Cos.
Grease extractor	Standard Steam Specialty Co.
Hot water heater	The Griscom-Russell Co.
Temperature control apparatus	Johnson Service Co.
Back pressure valve	H.S.B.W.-Cochrane Corp.
Exhaust heads	The Swartwout Co.
Traps	Crane Tilt-Crane Co.
CO_2 recorders	Republic Flow Meters Co.
Air compressor	Westinghouse Elec. & Mfg. Co.
Sump pump	Rumsey Pump Co.
Ejector	(Metropolitan) Manning, Maxwell & Moore Co.
Valves:	
Globe	The Lunkenheimer Co.
Gate	The Chapman Valve Mfg. Co.
Check	The Chapman Valve Mfg. Co.
Regulating valves	Mason Regulator Co.
Feed water filter	Ross Heater & Mfg. Co.
Flow meter	Republic Flow Meters Co.
Mechanical lubricators	Madison-Kipp Lubricator Co.
Suction strainer	Boylston Steam Specialty Co.
Ventilating fan and motor	Ilg Elec. Ventilating Co.

Some Notes on Centrifugal Pumps

WORKING DATA ARE PRESENTED ON INSTALLATION AND OPERATION. BY C. C. BROWN.

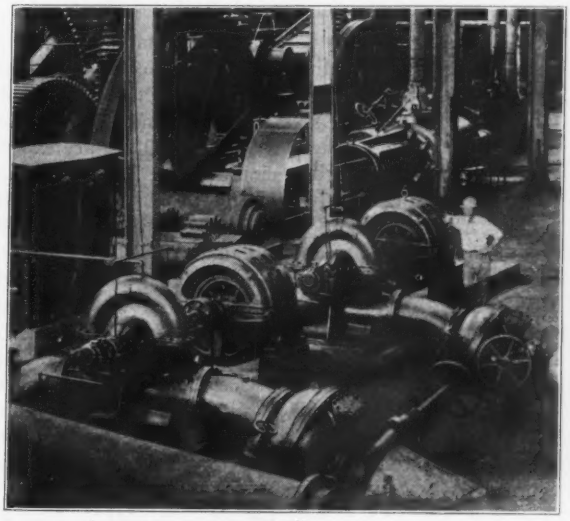
OF LATE YEARS, the centrifugal pump has been coming into its own in industrial plants and has to a great degree replaced the steam pump in sugar refinery practice. In one of the Western cane sugar refineries, where some 300 centrifugal pumps are in operation varying in size from 50 to 3500 g.p.m., a great deal of experience has been gained as to just what is the right design and construction of pumps for sugar refinery practice. While the points I am about to bring out apply directly to sugar refineries, they also apply to other industrial plants as well.

Since the infancy of the sugar refining game it has been almost universal practice to use steam reciprocating pumps for the pumping of water, lime saccarates, sugar juices, and in fact all liquids met with in this industry. One of the reasons for this condition is, undoubtedly, that inasmuch as steam, both live and exhaust, is one of the necessities for the present method of refining sugar, the latent energy released in dropping the pressure from say 100 lb. down to say 10 lb. may just as well be used to drive the pumps in the plant as to be reduced through a valve. The lower pressure is necessary for the operation of the evaporators.

Now that the steam balance of such plants is receiving more consideration and study with the idea of fuel economy in mind, and the use of steam turbines is becoming more and more prevalent, it has become apparent that the use of steam-driven reciprocating pumps with their low efficiency is giving way to the use of the electrically-driven centrifugal pump. The chief advantages of the centrifugal pump over the other types of pumps are its simplicity, reliability and ease of operation and repair. Another important feature that is particularly advantageous in the sugar refinery is that the discharge is smooth and continuous, and free from shocks and pulsations. Due to this fact large and cumbersome foundations are unnecessary. A third advantage of this type pump over the reciprocating steam variety is the fact that the discharge from the centrifugal pump may be shut off by merely closing a valve in the discharge line without dangerous pressures being introduced or requiring the motor to be shut down.

At the plant in question there was pumped some 10,000,000 gal. of salt water through the condensors or pans and evaporators per day. This water was lifted about 15 ft. from the ocean and pumped to the top of an eight-story plant, or against a head of 100 lb. Great difficulty was experienced in obtaining a combination of metals for pump casings and impellers that would stand up under this service. The first combination that was tried consisted of cast-iron casings, bronze impellers and steel shafts. It was soon found, however, that due to the action of the salt water, severe corrosion set in between the cast-iron casings and the bronze impellers, which resulted in the impellers being eaten away and the cast-iron casings rusted through. The next combination that was tried was all bronze casings and bronze impellers on steel shafts. Here again trouble was experienced in the form of pitting and corrosion of the steel shafts. New steel shafts were then installed composed of 3 per cent nickel steel, but still the trouble continued. A perfect combination was finally reached by using bronze casings and runners with 3 per cent nickel steel shafts upon which were placed bronze bushings, thus presenting an all bronze surface to the salt water. This combination, while perfectly satisfactory, was costly. As a final compromise, a combination was installed consisting of cast-iron casings, cast-iron impellers and 3 per cent nickel steel shafts with a renewal of the shafts and impellers once a year.

Another service where difficulty was encountered was the handling of hot syrups and sugar solutions. Here, again, many combinations were tried; cast-iron casings and cast-iron impellers on steel shafts were a decided



CENTRIFUGAL PUMP INSTALLATION IN A WESTERN SUGAR REFINERY

failure; cast-iron casings and bronze impellers on steel shafts were likewise unsatisfactory, the same corrosive and erosive action of the hot syrups upon the impellers being noted as in the case of the salt water pumps. In addition to that trouble, there was the excessive wear of the gland packing upon the shaft; it was necessary to keep the glands tight so that no sugar solution would get into the pump bearings. Upon several occasions when the sugar solution did get into the bearings, the latter were burned out in less than one-half hour. The combination which finally was adopted as being the most satisfactory was bronze casing, bronze impeller and monel metal shaft. Small water jets were connected to the glands so that a small stream of water was at all times flowing through the packing glands into the suction of the pump, thus preventing the sugar solution from getting out through the packing and into the bearings.

The only combination that was found to be satisfactory for use in pumping lime water or milk of lime was a pump consisting of white iron casing and white iron impeller upon a monel metal shaft. Thus it is that each

individual installation has its own problems of design and construction.

Installing Centrifugal Pumps

Aside from the problem of securing pumps which will stand up under particular service conditions there are also the important problems of correct installation and operation. Many of the operating difficulties are directly traceable to a lack of consideration for the few simple points connected with setting up centrifugal pumps.

Set the base plate with pump and prime mover on a solid foundation. Massive foundations are not necessary, but they must be sufficiently heavy to hold against belt strain, vibration, weight of pipes, etc., and must be homogeneous, so that they will support the pump base at all points. Foundations may be constructed of wood, brick or concrete. It has been found advantageous in many plants to construct the base of concrete, with a curb 3 by 3 in. around the edge to act as a basin, a drain pipe having been installed at the lowest point of the basin.

The combined base for pump and motor should be carefully leveled and lined up by means of wedges, so that the shaft turns easily by hand after the anchor bolts have been tightened. A straight-edge should be used to ascertain that the two halves of the coupling are exactly central. The straight-edge should be used all around the rim and over again after giving the coupling quarter turns. A caliper should show the coupling faces equal distances apart all around and in any position of shafts. Shafts of pump and prime mover must turn perfectly free. In bolting the pump down to the foundation, care must be taken not to spring the base which, if done, will cause the bearing to be thrown out of alinement, cramping the shaft, causing undue friction, heating with consequent loss of power and possible damage. The alinement should again be checked after the foundation bolts have been tightened and the suction and discharge lines connected.

Place the pump as near the water supply as possible, for the larger sizes, not exceeding 20 ft. and the smaller sizes 15 ft.; and where the suction line is long 10 ft. above the lowest point to which the water recedes when pumping. The suction lift including friction, varies with the size of the pump and the liquids to be handled. Hot water, such as boiler feed water, or heavy liquids, must flow to the pump under a positive head, this head varying according to the difference in temperature of the liquid or the viscosity of the liquid.

Suction Lines Must be Air-Tight

Suction lines must be absolutely air-tight, as the slightest amount of air will impair the working and produce no discharge. Avoid air pockets in the suction and discharge lines as air will collect in them and decrease the effective area of the lines. Wherever possible use long radius bends in place of elbows in both the suction and discharge lines. There should always be a continuous rise in the suction pipe toward the pump.

Never use pipe smaller than the flanges of the pump, and for long lines the discharge line should be one or more sizes larger than the pump discharge. A check valve should be provided on the discharge in all cases where the discharge line is of considerable length. The check valve and gate valve should be placed as close to the pump as possible, the check being placed between the gate valve and the pump in order that the check valve may be inspected or repaired without emptying the discharge line. Suction and discharge lines should be self-supporting or supported outside of the pump in order to avoid strain on the pump connections and casing and also to insure proper alinement.

Foot valves, where used, are fitted to the inlet of the suction pipe, and their free area should be considerably in excess of the area of the suction pipe. Unless means are provided for priming, all pumps having a suction lift should be provided with foot valves. Whether a foot valve is used or not, a strainer should be attached to the end of the suction line to prevent large substances from entering, and clogging the pump. Manufacturers do not as a general thing recommend foot valves for the reason that water hammer is easily caused by the use of these valves, causing breaks of important parts of the pump. To avoid this, they prefer the application of either a check valve or a gate valve bolted to the discharge of the pump with priming accomplished by means of either a hand or power primer, which exhausts the air from the pump, thus causing the water to rise in the suction pipe and completely fill the pump. Priming connections should always be located at the highest point of the pump casing.

Methods of Priming

There are several methods that may be used for priming centrifugal pumps. One method is to fit the pump with a steam ejector. The discharge valve being closed, the steam inlet valve to the ejector being first opened, and then the valve between the ejector and the pump opened, the air in the pump and suction line will be exhausted, and the water drawn up into them. When the ejector is placed near the pump, complete priming will be indicated by water issuing from the ejector. In shutting off the ejector, close the valve between it and the pump first, and the steam inlet valve last. Where it is not convenient to place the ejector near the pump, the air pipe may be extended in which case it is necessary that a slightly larger air pipe be used than when the ejector is placed near the pump.

Another method is to use a check valve in place of the discharge valve and a hand pump or power vacuum pump in place of the steam ejector. The priming being accomplished in the same manner as explained in the first method, there should be a valve placed in the air pipe which should be closed before starting. For hand priming a Douglas type pump may be used, piped with the air pipe forming a loop a little above the discharge. It is only necessary to put a little water into this style of pump to waterseal it, and make of it a good vacuum pump, the loop referred to above preventing the water from escaping. When a power driven vacuum pump is used for priming, a water trap or other means should be placed in the air pipe to prevent water entering and possibly breaking the primer. Where the valves on the vacuum pump are large, this may be omitted. On large pumps, a water glass similar to those used on boilers, placed near the top of the pump shell, will show when the priming is complete.

Still another method of priming consists of using a foot valve, in which case the pump and suction pipe are to be filled with water through the discharge or the top of

the pump casing, from any convenient source, such as a small tank or hand pump, which can be piped to the top of the centrifugal pump. Where the suction pipe is long, there should be at least 5 ft. of a discharge head on the pump to prevent the water being thrown out of the runner or impeller, before the water in the suction pipe begins to move, and thus cause failure to start. It is well to turn the runner around once or twice by hand to insure getting all of the air out of the arms, especially in small pumps. With vertical pumps where check or discharge valves are used a vacuum gage placed on the air priming pipe at the head of the well or pit will show when the pump is primed and avoid the necessity of climbing down into the well or pit. A vacuum gage may be used in the other methods described but care must be taken to shut off the gage before starting the pump, as pressure will ruin a vacuum gage. Where a pressure of water of 40 or 50 lb. is available, an air ejector may be operated for priming, substituting water for steam. This, however, requires a special ejector.

Operating Centrifugal Pumps

After erection, and before starting the pump, the bearings must be carefully cleaned with gasoline, and the oil reservoir filled with high grade lubricating oil as high as the top of the oil gage indicates. Ring oil bearings do not require oil cups, but the oil reservoir should be filled with oil to a height at which the oil rings will dip into it, but it is not a good thing to put too much oil into a ring oil bearing.

To avoid corrosion, the packing is removed from the stuffing-boxes before the pump leaves the factory. When repacking, the glands should be filled with soft packing, and on no account should the glands be drawn up tight, as this is sure to cause heating, and where the speed is high it will burn the packing and score the shaft. Pumps fitted with water sealed stuffing-boxes have a pipe and cock connecting the discharge case with the stuffing-box. This cock should be opened for water sealing purposes, but remember that if the water contains sand it is not wise to try to water seal at all.

The pump must always be primed before starting, otherwise the interior parts which depend upon the water for lubrication, will be injured. While priming, the pump must remain stationary, and should never under any condition be operated without water. In other words, it should never be allowed to run dry, or empty. As soon as the pump is primed, that is, completely filled with water, it is ready to start. It should now be started with the discharge valve closed. The pump will attain its normal speed in about 20 sec., after which the gate valve on the discharge should be gradually opened until full load is reached. Remember that, when starting with a closed gate valve on the discharge, the load on the motor is less than when the gate valve is wide open. As long as the water, and consequently the case, do not heat up excessively, due to friction produced by the rotation of the impeller the centrifugal pump may be operated with a closed discharge valve. In contrast with displacement pumps, no bypasses are required, nor can the pump or the piping system be damaged, as the shut-off or static pressure is only 10 or 15 per cent greater than the pressure at full capacity.

If after starting the pump it throws a little water at the first few revolutions, and then churns and fails to discharge more, it is evidence that all of the air was not expelled from the pump and piping or the suction lift is too great, or there is a leak in the suction piping or an insufficient discharge head. Pumps with horizontal over discharge and short discharge pipes are sometimes difficult to start owing to the fact that the water is thrown out of the pump before the water in the suction pipe is fairly started. The remedy for this is to add a length of discharge pipe.

If when first started the pump throws a full stream for a few minutes, and then fails, it is caused by failure of supply, or water in the well or pit receding below the suction limit, which in a well is best determined by a vacuum gage placed on the suction elbow of the pump. The remedy for this is to lower the pump, thus reducing the suction lift.

If the pump delivers a full stream of water at the surface, or level of the pump, but fails to pump at a higher discharge point, the speed of the pump is too low.

If the pump starts a full stream, and then the discharge decreases slowly until the pump fails to deliver any water, it is caused by a leak in the packing gland.

If when the pump is started it delivers and continues to deliver only a small stream of water there is probably one or more air pockets wherein air has collected in such quantity as to decrease the effective area of the line.

If the pump delivers a full quantity for a few hours and then fails, the speed and water supply being unchanged, the suction pipe or impeller is obstructed.

If when running there is heavy vibration, the shaft has been sprung, the pump is out of alinement, or an obstruction has lodged in one side of the impeller.

If the bearings heat unduly, the belt is unnecessarily tight, the bearings lack oil or there is an end thrust. Never use belts of greater length, thickness or width than is necessary to transmit the power required to drive the pump. Belts should be light and flexible.

In cases where it is necessary to operate the pump at a lower head than that for which it was designed or speeded, the gate valve on the discharge may be used to throttle the delivery and prevent pumping an excessive amount of water, which generally means an overload. Where a gate valve is not provided, often a piece of sheet steel with a hole in the center, properly proportioned, will serve as well. It is advisable not to operate pumps under excess conditions of capacity, as their efficiency will be much impaired and the cost of maintenance and upkeep will be increased.

Production of soft coal according to a recent government report continues at the rate of about 10,750,000 T. per week. The output, including lignite and coal coked, local sales, and mine fuel, in the week ended July 21 decreased somewhat and preliminary estimates place the total at 10,673,000 net tons, against 10,941,000 tons in the week preceding.

The rate of soft-coal production in 1923, as measured by the average daily production each week, has been remarkably uniform, ranging for the most part from 1,700,000 to 1,800,000 T. per working day. The two pronounced peaks for 1923 mark the holiday periods, New Year's Day and July 4, when complete shutdown of the mines on those days gave opportunity for improved car service during a few following days.

Insulation of Cold Surfaces to Prevent Sweating*

THICKNESS OF INSULATION TO PREVENT SWEATING MAY BE MATHEMATICALLY DETERMINED FOR VARIOUS CONDITIONS OF TEMPERATURE AND HUMIDITY. BY L. L. BARRETT

IT IS THE common practice to insulate cold surfaces with hair felt, cork, or similar substances for the purpose of preventing sweating even when the question of the heat gained by such surfaces is of no economic importance. Such insulations are frequently applied without proper consideration of the conditions which will surround their use and in such cases they often fail to give satisfactory service. Sweating may occur on the surface of the insulation due to insufficient thickness or it may occur on the cold surface itself due to

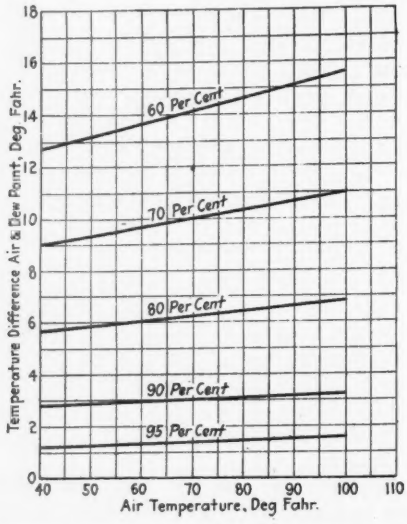


FIG. 1. TEMPERATURE DIFFERENCE BETWEEN AIR AND DEW POINT AT DIFFERENT RELATIVE HUMIDITIES, PLOTTED AS A FUNCTION OF AIR TEMPERATURE

the fact that the insulation is not air tight. A mathematical basis for determining the thickness of insulation required to prevent sweating on the surface of the insulation under various conditions of temperature and humidity is derived. Curves are presented which will obviate the necessity of any mathematical work in making practical determinations.

Moisture will be deposited on a surface whenever its temperature falls to that of the dew point. Figure 1 gives the temperature difference between the air and the dew point corresponding to different air temperatures and different relative humidities as computed from Goodenough's tables.

MATHEMATICAL ANALYSIS

From equations of heat flow from the warm air to the outer surface of the insulating covering, through the covering, and from the inner surface of the insulation to the cold medium, the author derives an equation

$$t = k \div c\,[(T^0 - T'') \div (T^0 - T') - 1] - k \div c'$$

where t = thickness of insulation required to prevent sweating, in.

k = co-efficient of heat conductivity of the insulation, B.t.u. per sq. ft. per hr. per in. thick per deg. difference in temperature between the inner and outer surfaces.

c = surface transmission co-efficient on warm side of insulation.

c′ = surface transmission co-efficient on cold side.

T^0 = air temperature.

T′ = temperature of the dew point.

T″ = temperature of the cold medium.

When the value of c′ is great, as is the case where there is a high air velocity on the cold side of the insulation, or when the cold surface of the insulation is separated only by a metal wall from a cold liquid such as cold water or brine, T″ becomes practically equal to T′, the surface temperature or the cold side of the insulation, the quantity (k ÷ c′) becomes negligible and the equation becomes

$$t = (k \div c)\,[(T^0 - T'') \div (T^0 - T') - 1] \qquad (2)$$

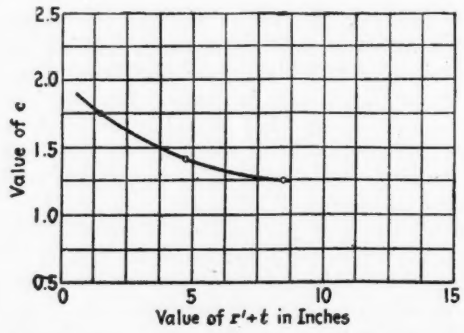


FIG. 2. TRANSMISSION CO-EFFICIENT FROM AIR TO COVERING AS A FUNCTION OF THE OUTER RADIUS OF THE COVERING

Temperature difference between the pipe and air at which sweating will occur, when the thickness of the covering and the temperature difference between the air and dew point are known is given by the equation:

$$T^0 - T'' = (T^0 - T')\,[c \div k\,(r + t)\log_e \{(r + t) \div r\} + 1] \qquad (3)$$

where r is the radius of the pipe.

VALUES OF c AND k

Heilman[1] has experimentally established the curves of heat loss per square foot of canvas covered surface of insulation as a function of $T^0 - T'$ in the case of pipes having diameters of the outer surface of the covering of 3.1 in., 9.5 in., and 17.2 in. respectively. The values of c given by these curves for a temperature difference of 10 deg., and which will not vary materially for temperature differences ($T^0 - T'$) from 0 to 20 deg. are: 1.74 for 3.1 in., 1.4 for 9.5 in., 1.25 for 17.2 in. These values have been used in Fig. 2 to establish a curve giving values for c as a function of r + t.

Values of k for the various coverings in common use, as determined by Van Dusen[2] are given in the table.

* Abstract of paper presented at the semi-annual convention of the American Society of Heating and Ventilating Engineers at Chicago, May 21 to 23.

[1] Trans. A. S. M. E., 1922.

[2] Journal Am. Soc. Heat.-Vent. Engineers, Vol. 26, page 646.

A value of 0.3 for k has been used in the computation of the curves given. As the thickness of covering

VALUES OF K FOR COMMONLY USED COVERINGS

	k	Per cent variation from k = 0.3
Hair Felt	0.246	—18
Keystone Hair	0.27	—10
Wood Felt	0.36	+20
Cork, 6.9 lb. per cu. ft.	0.27	—10
Cork, 9.7 lb. per cu. ft.	0.30	0
Cork, 16 lb. per cu. ft.	0.35	+17

required varies directly with the value of k for the covering used in the case of flat surfaces, and approximately so in the case of pipes, the thickness determined from the curves should be increased or decreased, according as to whether the value of k is greater or less than 0.3, and by the percentage indicated in the last column of the table shown above.

Discussion of Curves

Curves in Fig. 3 are plotted from equations 2 and 3, using the values of $T^0 - T'$ corresponding to the assumed humidities as given in Fig. 1 for an air temperature of 80 deg. F. It will be noted from Fig. 1 that the value of $T^0 - T'$ corresponding to any relative humidity varies but slightly with the air temperature for the air temperatures ordinarily encountered on the warm side of the insulation when sweating occurs. This makes it possible to disregard the air temperature in Fig. 3 and to label the curves with the humidities instead of with the corresponding values of $T^0 - T'$.

Curves are given for three different pipe sizes, for the greater the pipe size, the thickness of insulation and all other conditions remaining constant, the less will be the temperature of the surface of the covering, and the greater will be the insulation thickness necessary to keep the surface temperature above the dew point. The curves are straight lines in the case of the flat surface.

None of the curves pass through the origin because no insulation is required when the temperature difference between cold surface and the air is less than the temperature difference between the air and the dew point. For instance, at humidity 80 per cent the temperature difference between the air and dew point is 6.5 deg., therefore no insulation is required until the temperature difference between the cold surface and the air is equal to 6.5 deg.

Example of Method of Computation of Curves

Let it be required to determine a point on the 95 per cent humidity curve for 3-in. pipe and assuming a covering 1 in. thick.

$(r + t) \log_e [(r + t) \div r)] = 2.75 \log_e (2.75 \div 1.75) = 1.24$

$k = 0.3$

$c = 1.6$ [from Fig. 2, (for $r + t = 2.75$)]

$T^0 - T' = 1.5$ deg. (from Fig. 1)

$T^0 T'' = 1.5 [(1.6 \div 0.3) (1.24) + 1] = 11.4$

Experiments of Heilman, from which the values of c are obtained, were conducted under still air conditions, that is, where the air circulates due to natural convection currents only. If the air on the warm side of the insulation have sufficient velocity the values of c will be considerably greater than the values given in Fig. 2, and the greater the value of c, the less will be the thickness of insulation required. Therefore the thicknesses given by the curves are maximum and are applicable to still air conditions, as where the cold surfaces are located in confined spaces. It has seemed best to prepare the curves in Fig. 3 for still air conditions, as this condition gives the best standard of reference, and at the same time gives the worst conditions as regards the prevention of sweating. Furthermore, the values of c corresponding to this condition are well established by experiment.

If the cold surfaces are located where there is much air circulation on the warm side of the insulation, the thicknesses shown in Fig. 3 may be reduced. The experiments of T. S. Taylor[3] have shown that the value of c increases approximately uniformly with the velocity at low velocities and that with an air movement of 800 ft. per min. the value of c will be 5.71. The value of c for

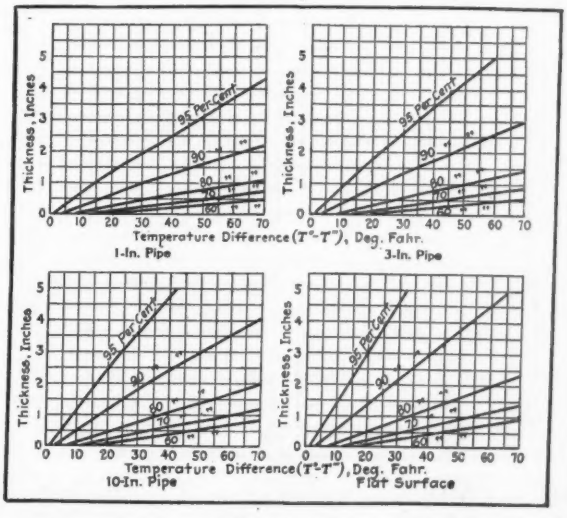


FIG. 3. THICKNESS OF INSULATION NECESSARY TO PREVENT SWEATING AT DIFFERENT RELATIVE HUMIDITIES, PLOTTED AS A FUNCTON OF TEMPERATURE DIFFERENCE BETWEEN COLD MEDIUM AND AIR

still air found in the same determinations at a temperature difference of 10 deg. F. is 1.68. Since the thickness of insulation required for flat surfaces varies inversely as the value of c, it is a simple matter to utilize this data to establish a curve showing the percentage reduction for different air velocities to be applied to the values of t given in Fig. 3. This curve is given in Fig. 4. For practical purposes this curve can also be used in the case of pipes without any serious error.

It should be pointed out that insulation should be made with caution, and in most cases should not be made at all. Still air conditions will usually prevail at one time or another, and unless the insulation thickness required by such conditions is applied sweating will occur.

Insulation of Roofs and Exterior Walls

When insulation is applied to roofs or exterior walls to prevent sweating the maximum effect of outside air

[3] Journal American Society Heating and Ventilating Engineers, Vol. 26, page 451.

velocity is to reduce the temperature of the cold side of the insulation T′′ to the temperature of the outside air. Equation 2 is then applicable and the flat surface curves in Fig. 3 give the proper thickness of insulation. For lesser outside air velocities the thickness of insulation could be somewhat reduced, but due to the fact that high outside air velocities occur from time to time it is advisable to consider T′′ equal to the temperature of the outside air. This will give a thickness suitable for the most extreme velocities. The thickness required may also be reduced according to the insulating effect of the structural materials entering into the construction of the wall or roof. For instance, if the wall is faced with 8 in. of brickwork of conductivity k = 5, the thickness of insulation given by the curves may be reduced by $0.3 \times 8 \div 5 = 0.48$ in.

Insulation of Cold Spaces

When it is desired to insulate cold spaces containing air at a definite low temperature, such as refrigeration

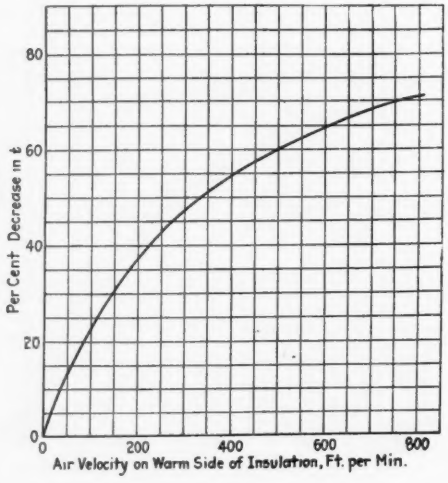


FIG. 4. PERCENTAGE DECREASE IN THICKNESS FOR INCREASING AIR VELOCITIES. BASED ON THICKNESS GIVEN IN FIG. 3 FOR FLAT SURFACES

rooms, in such a way that there shall be no condensation on the warm side of the insulation, the surface transmission must be considered for the cold side as well as for the warm side of such insulation. In such cases equation 1 is applicable. The thickness of insulation given by this equation is less than that given by equation 2 by $k \div c'$, which is equal to 0.24 in. when $k = 0.3$ and $c' = 1.25$. The flat surface curves in Fig. 3 may therefore be used to determine the insulation thickness necessary to prevent sweating in the case of such cold spaces, 0.24 in. being subtracted from the thickness given by the curves to obtain the final result. For conductivities other than 0.3 it will be noted from equation 1 that the thickness required will vary directly with the conductivity. The thickness of insulation may also be reduced according to the insulating effect of the other materials entering into the construction of the walls of such cold space.

Use of the Curves

Variables to be considered in the use of these curves are temperature difference (T^o—T''') between air on the warm side of the insulation and the cold medium, relative humidity, pipe size, conductivity of insulation, and amount of air circulation. The maximum thickness of insulation will be required when the temperature difference T^o—T''' is the greatest, when the humidity is the highest, when the conductivity is the greatest, and when the air circulation on the warm side of the insulation is the least.

Humidity is the greatest and the most important variable. The following observations may assist in determining the humidity to be allowed for in structures under design. In the northeastern states the relative humidity of the outside air during the winter months varies generally between 30 and 100 per cent. The air within heated buildings which are not equipped with any means of humidification will vary generally during these months from 15 to 35 per cent. Where the buildings are equipped with humidification systems for the purpose of maintaining bodily health and comfort the humidity will vary from 40 to 50 per cent. During the summer months also the humidity of the outside air will vary generally between 30 and 100 per cent, but the humidity inside of buildings will run slightly higher than that of the outside air, except in the case of factories where heat is being used or generated, where the humidity will be less. During fogs the air is saturated and the humidity is 100 per cent.

Choice of Insulation

After arriving at the correct thickness to prevent sweating on the outer surface of the insulation the next important point is to choose an insulation that is as nearly air tight as possible. If any air reaches the cold surface through cracks or joints in the insulation moisture will be deposited on the surface. The insulation will then become gradually water soaked from the inside and rendered worthless as an insulator. Cork appears to be the most suitable material for the purpose, as it lends itself readily to the attainment of an air tight job. All joints and cracks can be effectively sealed with brine putty. When other materials are used, special effort should be employed to protect them from moisture and to obtain a minimum of infiltration.

Considerable interest is centered in the problem of developing the waters of the Colorado River for irrigation and power and to lessen danger from floods in Imperial Valley. The first thing needed in connection with any such development is a survey, but a 300-mi. stretch of this 1500-mi. river, including the ruggedest part of the Grand Canyon, has not yet been surveyed in any detail. The surveying and mapping of this stretch, which includes the dangerous gorges of the Marble and Grand Canyons, was started on Aug. 1, as was announced at the Department of the Interior recently. This part of the river's course, which is crowded with bad rapids that swirl between steep rock banks, has been traversed on only six previous occasions. It was first explored in 1869, by Major John W. Powell, later director of the Geological Survey. The present party of engineers and geologists of the Geological Survey will make a trip by boat from Lees Ferry through the canyons to the mouth of the Virgin River, in Arizona, a distance of about 300 mi., and will make records of the slope of this entire stretch of the river and of the topography.

Practical Construction of Alinements Charts---II*

DETAILED INSTRUCTIONS FOR CONSTRUCTING TYPICAL ALINEMENT CHART. BY R. T. LIVINGSTON†

IN THE PREVIOUS article which appeared in the Aug. 15 number of Power Plant Engineering, the advantages and methods of use of alinement charts were shown as well as an outline of the method of their construction. In order to show clearly the method and order of procedure and to indicate the problems which will be met and the method of solving those problems, a particular example will be taken and a chart designed in this article.

An approximate formula for allowable time of closure of water turbine gates with a given pressure for an installation with a pipe line is

$$t = \frac{2 \times L \times V}{g \times h}$$

where t = time in seconds
L = length of penstock in feet
v = velocity in feet per sec.
g = gravitational constant in feet per sec. per sec.
h = allowable pressure rise in feet

It is first necessary to transform the formula into the easiest form for solution, in two, or if necessary more, steps. It will be divided

$$(2 \times L \times V)\frac{1}{g \times h}$$

and the first problem is to design a chart which will solve the formula $(2 \times L \times V)$. There is some choice in the selection of factors at this point and it is often well to try and combine those factors which must be constant for any fixed case and to leave the other factors which will have a varying effect upon the answer to be solved afterwards, so that the effect of change in the factor can be more easily seen. The length of penstock therefore and velocity in penstock which would most likely remain constant for any one case are first taken.

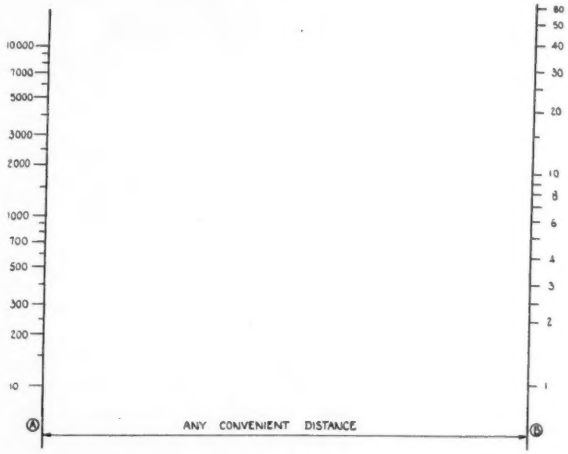


FIG. 1. FIRST STEPS, SHOWING MANNER OF LOCATING (A) AND (B) SCALES

L, the length of the penstock, is laid out on scale (A) from 100—10,000 ft., a double "log" scale. The V, on scale B was taken any convenient distance from A. Here it was assumed that velocities of from 1 ft. per sec. —25 ft. per sec. might be met in practice and it would have been possible to make the length of the two lines the same but to different log-scales, but this would have necessitated special logarithmic divisions and so one of the log scales that a strip for was at hand was chosen and hence the divisions run up as high as 60 ft.per sec. Figure 1 illustrates what has been obtained so far.

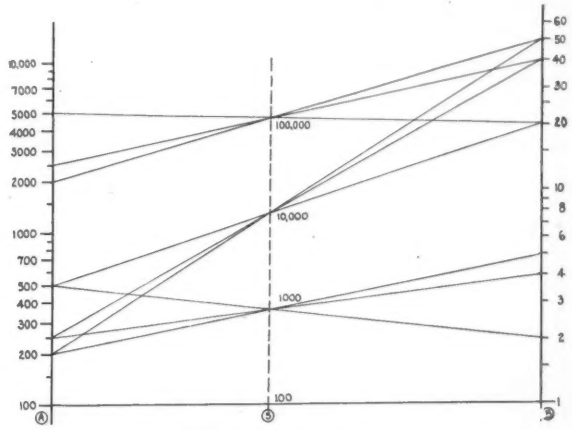


FIG. 2. METHOD OF LOCATING (S) SCALE

It is now desired to determine the position of the center or support line. In order to determine this, it is necessary to compute at least two points on it and it is better to compute three. Now in order definitely to locate a point we must have two, or better three intersecting lines. Let us take the numbers 1000, 10,000 and 100,000 as representing the product of $2 \times L \times V$. Now the following conditions satisfy our equation.

$2 \times L \times V$	L	V
1000	100	5
	250	2
	200	2.5
10,000	100	50
	250	20
	200	25
100,000	1000	50
	2500	20
	2000	25

We will now draw lines connecting the L's and V's for determination of the 1000 point. (See Fig. 2.) These lines intersect in a point which is the desired point. In a similar manner we determine the points 10,000 and 100,000. These three points having been determined, we connect them with a straight line the dashed line in Fig. 3. At this point, we have an excellent check upon the accuracy of our construction. These three points must not only be on the same vertical line but this line must be parallel to the two factor lines previously taken.

At this point it is well to ink in the three lines thus definitely fixed and to mark the divisions on the two

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† Hydro Mechanical Engineer, New York City.

outside lines at least. These scale divisions are traced directly through from the strips placed under the tracing cloth and having traced them, the strips are removed and are available for further use. In marking divisions, care should be taken to make them heavy enough but not too long. As a general rule, do not cross the line with them as sometimes scales are doubled up on one line. Also when placing figures alongside the scale, do not label too many divisions, it is seldom necessary to label any of the fives except 15 and in a large scale 25. It is a common mistake to put in more divisions than the accuracy of the chart warrants. Remember that the eye

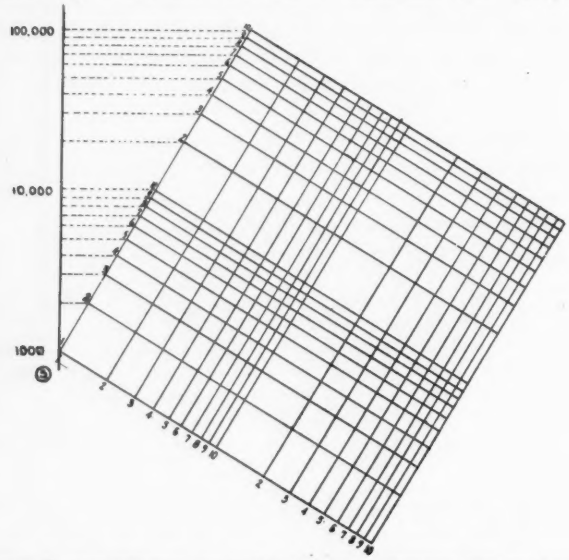


FIG. 3. PROJECTION OF LOGARITHMIC DIVISIONS WHEN LOG SCALE IS TOO LARGE

can divide any space into four parts at least as accurately as is necessary.

On the support line it is not always necessary to mark the divisions in ink. The line should be inked in and as many points as necessary located in pencil for further construction; but unless that particular product is wanted, there is nothing to be gained in inking in the divisions. Many people leave this line without any divisions, but the author has found it more convenient to mark it off in regular decimal divisions which can be used as reference points and nothing else. (See Figs. 3 and 4 in previous article.)

If this product, $2 \times L \times V$ was desired, however, it would be necessary to place a scale on it. Three points have been determined and it is necessary to run a log scale between these three points. The chances are that no log paper will be on hand which will exactly fit the conditions. If not, there are two possibilities: the smallest log scale may be too big, or the largest log scale too small. In the first case, it would be necessary to put the log paper under the tracing cloth so that the (1) point coincides with the 1000 on the support line; now turn the log paper at an angle, until the 10 point is on a horizontal line with the 10,000 point. Now project all the intersection of the log paper's horizontal divisions with its left-hand edge to the support line. In the second case, as before, make the 1 point coincide with the 1000 on the support line and make a point on the horizontal 10 line coincide with the 1000 on the support line and where the various horizontal log lines intersect the support line mark the divisions. The procedure in each case is illustrated respectively in Fig. 3 and Fig. 4.

This then fully determines the two factor lines A and B and the answer or support line S. It now remains to divide this answer by g . h, a constant times a variable.

Before going on, it would be well to consider in detail the effect of changes of scale and position of the lines. If for multiplying the B scale is made smaller, the scale on S is reduced, and if the B scale is made larger, it is in-

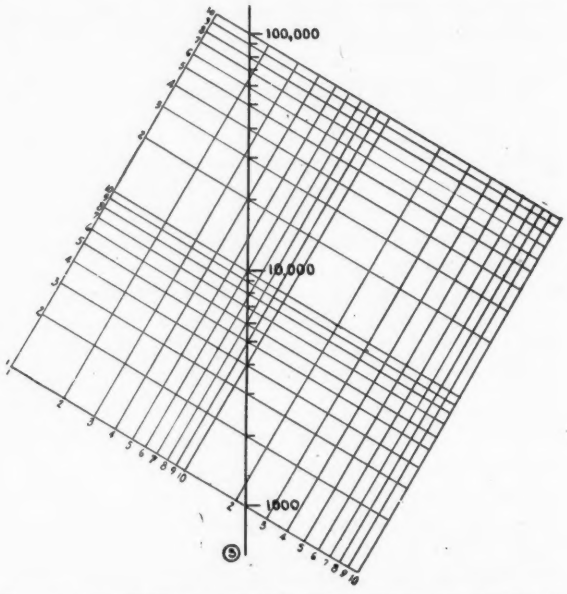


FIG. 4. MANNER OF DETERMINING DIVISIONS WHEN LOG SCALE IS TOO SMALL

creased. In some cases of division, it is desired to make the S scale quite large; this is accomplished by throwing it to the right of the B line. To throw it to the right, the scale on B must be reversed; that is to say, it is made to run the same as if multiplication were to be done. Of course if the two scales on A and B were the same length, then there could be no intersections, hence the B scale must be different from the A scale—usually larger. The ratio of the size of the two scales determines the position of the S line. This procedure is not recommended unless the numbers expected on the B scale need not be taken to great accuracy. Thus, if pipe sizes are desired, this would be perfectly feasible, for certain definite points would be desired; but if interpolation between these points were wanted, it would be a poor plan.

To continue: The next two scales; the h line and the final answer line, called the Y line and the Z line hereafter were desired inside the A and B lines. This position is not of necessity but merely choice. The same method as outlined for the determination of the S line is gone through for the determination of the Z line; but in this case, more trials and more care must be taken in the choosing of a position and scale for the Y line. In this case, any convenient scale is taken for the Y line that will make it approximately the same length as the (A) line. After three points on the Z line have been determined, the line is drawn and the divisions are marked in exactly the same way as shown for the S line.

When this is done we have the chart complete as shown in Fig. 3 of the previous article.

It is to be noted that in this case it is a simple division and that no odd powers enter into the calculation, hence it is a comparatively simple matter to settle the position of the Z scale.

Placing the various lines and the choice of the log scales is a matter of experience; but if one chart is made and various changes in the position of the lines and scales on the lines are tried, it will soon become quite simple. If factors are used whose values run between very narrow limits, it may be necessary to take the values of logarithms from a table and lay them out to some convenient scale, thus tenths, twentieths, fortieths, etc. Thus if one factor ran between 16 and 18, it would be necessary to lay the scale out to the hundredths and there is no logarithm paper that will do the work.

As was stated at the beginning of the previous article, no attempt has been made to show the theory of alinement charts nor the mathematical method of calculations; but it should be kept in mind that it is an application of the principles of logarithms and nothing more. By following this method, anyone who can use a slide rule, a drafting board and common sense can construct alinement charts.

In the next and last article appearing in an early issue, a number of more complicated equations will be solved and the reader, if interested, should attempt the solution of similar equations to demonstrate to his own satisfaction the various methods of procedure outlined.

Automatic Control for Oil Fired Boilers

SCHEME USED IN PLANT OF JOHN B. STETSON CO. HAS GIVEN SATISFACTORY RESULTS. BY S. B. DERBY*

CONTROL of the combustion in a steam boiler furnace consists essentially in regulating the quantity of fuel burned from instant to instant in accordance with the steam demand or load on the boiler. A change in steam demand cannot in most plants be anticipated before it occurs, therefore, the apparatus for regulating combustion is made as sensitive as possible in order to respond promptly to the first change in boiler steam pressure due to a variation in load. If, for example, the pressure falls below normal, corresponding to an increase in steam demand, additional fuel must be burned as quickly as possible, and the heat generated supplied to the boiler so that no appreciable drop in steam pressure will occur. If the pressure rises, due to a decrease in steam demand, the supply of fuel must be instantly lowered to avoid steam loss through the safety valves.

Oil fuel is well adapted for automatic combustion control, because it can be burned the instant it is introduced into the furnace. The burner for atomizing the oil, that is, for breaking it up into minute fog-like particles must, however, be designed to handle properly and efficiency widely varying quantities of oil. Each burner must be capable of the maximum variation in rate of feed required since there is no such thing as varying the number of burners in action.

The purpose of this article is to describe the automatic control developed by Mr. Harvey, chief engineer of the J. B. Stetson Co., of Philadelphia. The system has given quite satisfactory service and has many noteworthy features.

As shown by Fig. 1 the quantity of fuel oil passed to the burners is controlled by the boiler steam pressure. When the latter momentarily increases above the predetermined normal pressure, due to a diminution in steam consumption, the oil flow is shut-off. If subsequently there is a slight diminution in boiler pressure below the predetermined value, due to an increase in steam consumption, the oil is again supplied to the burners. In other words, the oil flow to the burners is repeatedly started up and shut down, corresponding to momentary variations of the boiler pressure below and above the normal operating pressure. In no instance is the flame issuing from any of the atomizers extinguished—enough oil being passed to the burner to keep the burner ignited at all times.

Arrangement for each boiler consists of a Spencer damper regulator diaphragm dome with fulcrum, lever

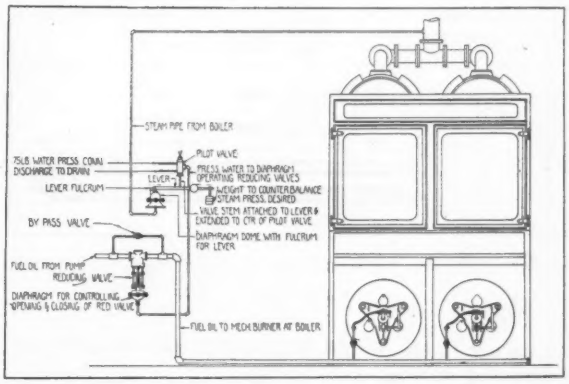

STEAM PIPE FROM BOILER
PILOT VALVE
75LB WATER PRESS COND DISCHARGE TO DRAIN
PRESS WATER TO DIAPHRAGM OPERATING REDUCING VALVES
LEVER
LEVER FULCRUM
WEIGHT TO COUNTERBALANCE STEAM PRESS. DESIRED
VALVE STEM ATTACHED TO LEVER & EXTENDED TO CTR OF PILOT VALVE
BY PASS VALVE
DIAPHRAGM DOME WITH FULCRUM FOR LEVER
FUEL OIL FROM PUMP
REDUCING VALVE
DIAPHRAGM FOR CONTROLLING OPENING & CLOSING OF RED VALVE
FUEL OIL TO MECH BURNER AT BOILER

FIG. 1. QUANTITY OF FUEL OIL PASSING TO THE BURNERS IS CONTROLLED BY THE BOILER STEAM PRESSURE

and weights counterbalancing the desired steam pressure and a Locke two-way pilot valve with a valve stem attached to and operated by the lever. This valve is a two-port valve whereby, when the pilot valve is in one position, water under 75 lb. pressure is transmitted to a diaphragm controlling a reducing valve in the oil line to the burner. When the pilot valve is in the other position, connection is made with the drain and the water pressure is relieved. The diaphragm referred to controls the opening and closing of a Schade reducing valve.

When the pressure of the boiler steam diminishes slightly below the normal value, the longer arm of the diaphragm dome lever falls, thereby lowering the valve stem of the water pilot valve. Under these conditions, the hydraulic pressure below the diaphragm controlling the reducing valve is relieved, and the valve is opened wide, allowing the oil to flow freely to the burners. On the other hand, when the pressure of the boiler steam momentarily increases, the long arm of the diaphragm dome lever is raised, in turn raising the valve stem of the pilot valve and transmitting the water pressure to

*Engineer, Schutte-Koerting Co., Philadelphia, Pa.

the reducing valve diaphragm, thereby causing the valve to close. A bypass around the latter permits enough oil to flow to the burner to keep the burner ignited when the valve is closed.

Burners are Schutte & Koerting type D mechanical fuel oil atomizers and are supplied with the air control registers. The burner, as shown in Fig. 2, consists of a burner body, clamp, tee handle, gooseneck, burner tube, guard tube and burner tip. The tip contains a spiral chamber formed by a triple thread screw. The screw is kept from rotating by a square shank secured in a socket in the base of the burner. The relation of the parts to each other may be clearly seen in the drawing.

Fuel oil for the burners coming from the storage tanks passes through a strainer, a pump whereby its pressure is raised to 100 to 250 lb. per sq. in., an oil heater wherein it is heated until thin enough to permit atomization (200—300 deg. F.), a second strainer, and the combination regulating valve. Here it passes in parallel flow through the bypass and through the main valve when the steam pressure is low, and through the bypass only when the steam pressure is high. From the regulator it is forced through the header, the burner tubes and around the spiral in the burner tips. The rotation imparted in the spiral causes the oil on leaving

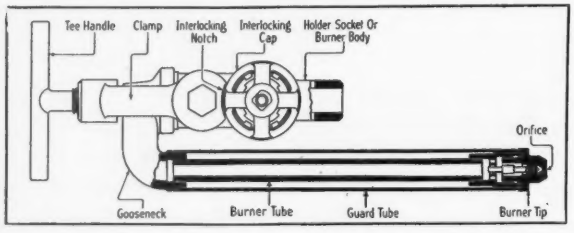


FIG. 2. BURNERS USED ARE OF THE MECHANICAL ATOMIZING TYPE

the burner to spread out into a wide cone of uniformly distributed oil fog.

Air is induced through the register where it is thoroughly mixed with the atomized oil, thereby insuring complete combustion of the fuel. The air register can be adjusted to supply the right amount of air for any rate of combustion.

Another feature of the burner for use in connection with the pressure regulator is its flexibility. Naturally, it is impossible to run at less than the least capacity of the burner or more than the greatest; that is, the regulator can only hold a constant steam pressure at ratings in between these extremes, but the latter are far enough apart to cover most practical operating requirements.

When the load passes beyond the capacity of the automatic regulator, one or more burners may be cut out by hand. As mechanical burners are always installed in smaller units than air or steam atomizers, the cutting out may be done in small steps, thus avoiding the necessity of running the remaining burners at full capacity. For a change in operating conditions which will be more or less permanent the burner tips may be changed and all the benefits of a well distributed heating effect may be obtained at low rates of combustion.

These burners will operate with any oil pressure between 75 and 200 lb. per sq. in. The increase in oil consumption with 200 lb. pressure over that with 75 lb. pressure varies from 33 per cent with 1 mm. burner tip to 50 per cent with the 2.75 mm. burner tip.

This equipment is installed in the plant of the John B. Stetson Co., Philadelphia, Pa., where it operates 29 mechanical fuel oil atomizers for a total of 4250 b.hp. Parker boilers. The regulation usually obtained is within one-quarter of a pound.

Failure of Steam Gage Pipe Causes Some Excitement

By H. J. Cheetham

ONE of the practices on navy ships having water tube boilers is the holding of what is known as a "leaky tube drill." The drill provides every man in the fire room with a special duty in the event of a tube bursting. Air lock doors in closed fire room jobs are closed, and guards stationed inside and outside. Men are detailed to speed up the forced draft fans so there will be a large rush of air through the furnace, to carry the steam up the funnel and keep it out of the fire room, and to stand by the feed pumps and the fire quenchers.

While this is fine from a safety standpoint, there are times when it results in rather amusing experiences. On one occasion I have in mind, the ship of which I was then chief engineer was almost ready to leave her anchorage when a burst tube in the forward fire room was reported to me. Since the rules and regulations forbade me, or anyone else, to open a fire room door, until the steam pressure in the damaged boiler was less than 50 lb., I hurried down to the engine room and got in touch with the Petty Officer in charge of that fire room through the speaking tube.

He reported everything going absolutely on schedule. Every man at his post, the fire quenched, the boiler shut off the line, the steam all being carried up the funnel, no one scalded and steam falling rapidly. Soon he reported steam below 50 lb. Then the air locked doors were opened, and the chief machinist, the chief stoker and I hurried down to view the damage.

Everything looked mighty placid in that fire room, and the only evidence of trouble seemed to be some steam blowing out from the steel casing at the side of boiler by the top steam header. I looked at it more closely, and it appeared to come from the small copper steam pipe to the pressure gage, which here ran under the flange of an angle bar stiffener almost out of sight. I shut off the cock on the steam header and our "burst tube" vanished.

It was the pipe to the steam gage which had burst. The crestfallen look of that Petty Officer when he discovered that the perfect discipline and steadiness of his crew had been wasted on such a trifle was a picture to remember. The hustle that followed to get that boiler back on duty again, and the ship into her place in the flotilla without the Squadron Commander discovering what a blunder had nearly been committed, I think that fire room crew will not soon forget. But it was a light, quick-steaming Yarrow boiler, and that "Brass Hat" never knew how nearly one of his ships had reported "unable to proceed to sea" because a pressure gage pipe had burst.

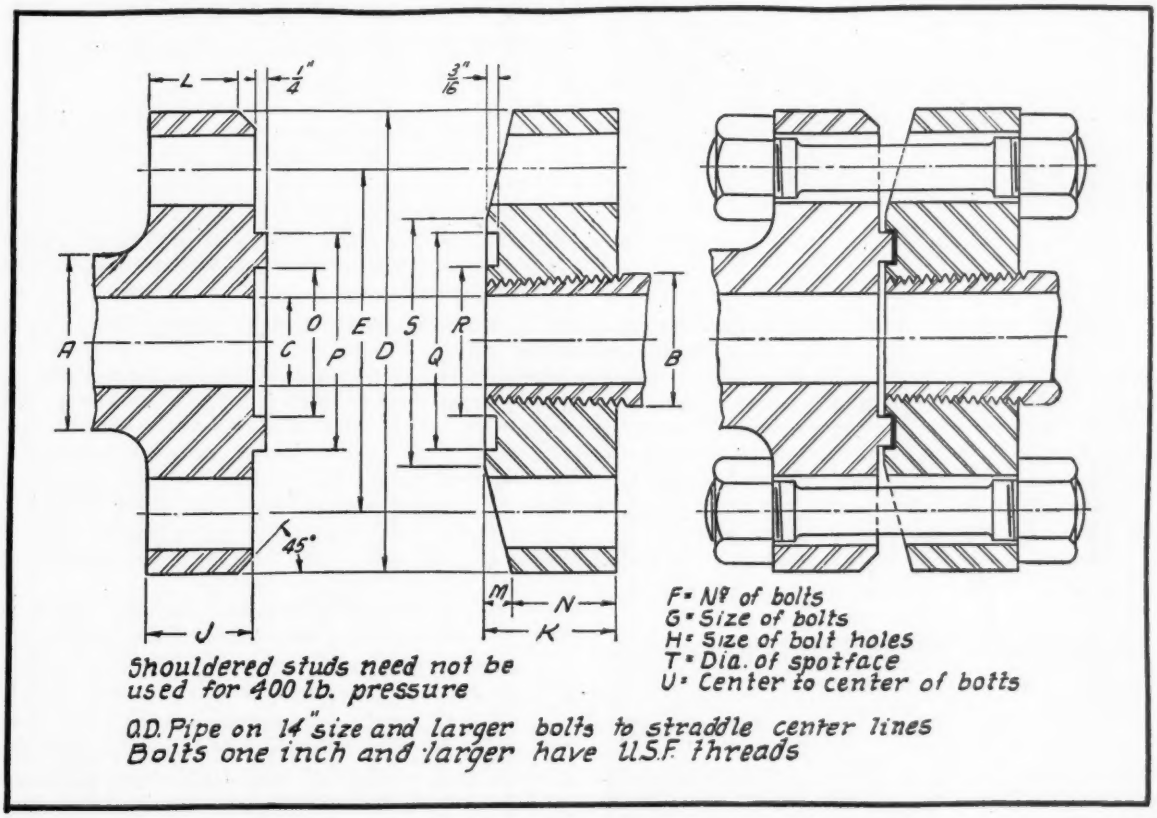
Standards for Tongue and Groove Joints

RECOMMENDATIONS BY OLAF E. OLESON FOR THE STANDARDIZATION OF FLANGES AND BOLTING FOR STEAM SERVICE

IN THE July 15 issue of Power Plant Engineering, a general discussion and tabular information were published on the recommendations for standards for flanges and bolting for the Sargol type of joint as applied to pressures ranging from 400 to 1200 lb., inclusive.

Tables given herewith are the recommended practice for the tongue and groove type of joint for pressures of 400, 500 and 600 lb. As for the Sargol joint, it is recommended that the present American standard for extra heavy drilling be extended up to and including 500 lb.

TABLE I. DIMENSION TABLE RECOMMENDED FOR AMERICAN EXTRA HEAVY 400-LB. PRESSURE STANDARD, TONGUE AND GROOVE JOINT



Pipe Size	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P	Q	R	S	T	U	Size
½	1 1/16	27/32	9/16	3½	2¼	4	7/16	½	¾	1	½	3/16	13/16	13/16	1 7/16	1½	¾	1¾	1⅜	1.94	½
¾	1 3/8	1 1/16	¾	4	2⅞	4	7/16	½	¾	1	9/16	3/16	13/16	1 5/16	1 15/16	2	1¼	2⅜	1⅜	2.03	¾
1	1 13/32	1 5/16	31/32	4½	3¼	4	½	⅝	13/16	1 1/16	⅝	3/16	⅞	1½	2¼	2 5/16	1 7/16	2⅝	1⅝	2.29	1
1¼	2 1/32	1 21/32	1 9/32	5	3¾	4	½	⅝	⅞	1⅛	11/16	3/16	15/16	1 13/16	2 11/16	2¾	1¾	3	1⅝	2.63	1¼
1½	2⅜	1 29/32	1½	6	4½	4	⅝	¾	1	1¼	¾	3/16	1 1/16	2 3/16	3 1/16	3⅛	2⅛	3⅝	1¾	3.18	1½
2	2 15/16	2⅜	1 15/16	6¾	5	4	¾	⅞	1⅛	1⅜	13/16	3/16	1 3/16	2 11/16	3 3/16	3⅝	2⅝	4	2	3.53	2
2½	3 5/16	2⅞	2 5/16	7¾	5⅞	4	⅞	1	1¼	1½	15/16	3/16	1 5/16	3 3/16	4 3/16	4¼	3⅛	4⅞	2⅜	4.15	2½
3	4 1/16	3½	2 15/16	8¼	6⅝	8	¾	⅞	1 7/16	1¾	1 3/16	¼	1½	3 13/16	4 13/16	4⅞	3¾	5¾	2	2.53	3
3½	4⅝	4	3⅜	9	7¼	8	¾	⅞	1½	1⅞	1 5/16	5/16	1 5/16	4 5/16	5 5/16	5⅜	4¼	6⅜	2	2.77	3½
4	5 1/16	4½	3 13/16	10	7⅞	8	⅞	1	1⅝	2	1 3/16	5/16	1 11/16	4 13/16	5 13/16	5⅞	4¾	6⅞	2⅜	3.02	4
4½	5 5/16	5	4 5/16	10½	8½	8	⅞	1	1 11/16	2⅛	1 5/16	3/8	1¾	5 5/16	6 5/16	6⅜	5¼	7½	2⅜	3.25	4½
5	6 5/16	5 5/16	4 13/16	11½	9¼	8	1	1⅛	1¾	2¼	1 5/16	7/16	1 13/16	6 1/16	7 1/16	7⅛	6	8⅛	2¾	3.54	5
6	7¼	6⅝	5¾	12½	10⅝	12	⅞	1	1⅞	2⅜	1 9/16	7/16	1 15/16	6 15/16	7 15/16	8	6⅞	9⅜	2⅝	2.75	6
7	8¼	7⅝	6⅝	14	11⅞	12	1	1⅛	2	2½	1½	7/16	2 1/16	8 1/16	9 1/16	9⅛	8	10¾	2¾	3.08	7
8	9⅜	8⅝	7⅝	15½	13	12	1⅛	1¼	2⅛	2⅝	1⅝	7/16	2 1/16	9 1/16	10 7/16	10½	9⅛	11¾	3	3.36	8
10	11¾	10¾	9¾	17½	15¼	16	1⅛	1¼	2¼	2⅞	1⅞	9/16	2 5/16	11 5/16	12 13/16	12⅞	11¼	13⅞	3	2.98	10
12	14	12¾	11¾	20½	17¾	16	1¼	1⅜	2½	3⅛	1 15/16	9/16	2 5/16	13 7/16	14 15/16	15	13⅜	16¼	3⅛	3.46	12
14	15⅛	14	12⅞	23	20¼	20	1¼	1⅜	2⅞	3½	2 5/16	9/16	2 1/16	14 7/16	15 15/16	16	14⅜	19⅞	3⅛	3.17	14
15	16¼	15	13⅞	24½	21½	20	1¼	1⅜	3	3⅝	2 5/16	9/16	3 1/16	15 11/16	17 3/16	17¼	15⅝	20	3⅛	3.37	15
16	17¼	16	14¾	25½	22½	20	1⅜	1½	3⅛	3⅞	2½	11/16	3 3/16	16 13/16	18 7/16	18½	16¾	20⅞	3¼	3.52	16
18	19½	18	16¾	28	24¾	24	1⅜	1½	3¼	4	2½	11/16	3 5/16	18 13/16	20 1/16	20¾	18¾	23⅛	3¼	3.23	18
20	21¾	20	18¾	30½	27	24	1½	1⅝	3½	4¼	2 11/16	11/16	3 5/16	20 13/16	22 11/16	22⅝	20¾	25¼	4¼	3.53	20
22	23⅝	22	20½	33	29¼	24	1⅝	1¾	3¾	4⅝	2⅞	¾	3 13/16	23⅜	25⅜	25¼	23⅛	27⅜	4½	3.82	22
24	25¾	24	22½	36	32	24	1¾	1⅞	4	4 13/16	3 1/16	¾	4 1/16	25 1/16	27⅜	27¼	25	30	4¾	4.17	24

TABLE II. DIMENSION TABLE RECOMMENDED FOR AMERICAN EXTRA HEAVY 500-LB. PRESSURE STANDARD, TONGUE AND GROOVE JOINT

Pipe Size	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P	Q	R	S	T	U	Pipe Size
½	1 1/16	27/32	9/16	3 1/2	2 1/4	4	7/16	1/2	3/4	1	1/2	3/16	13/16	13/16	1 7/16	1 1/2	3/4	1 3/4	1 3/8	1.94	½
¾	1 3/8	1 1/16	¾	4	2 7/8	4	7/16	1/2	¾	1	9/16	3/16	13/16	1 5/16	1 15/16	2	1 1/4	2 1/8	1 3/8	2.03	¾
1	1 19/32	1 5/16	31/32	4 1/2	3 1/4	4	1/2	5/8	13/16	1 1/16	5/8	3/16	7/8	1 1/2	2 1/4	2 5/16	1 7/16	2 5/8	1 5/8	2.29	1
1 1/4	2 1/32	1 21/32	1 3/32	5 1/4	3 3/4	4	5/8	3/4	1	1 1/4	3/4	3/16	1 1/16	1 13/16	2 11/16	2 3/4	1 3/4	3	1 3/4	2.63	1 1/4
1 1/2	2 3/8	1 23/32	1 1/2	6 1/4	4 1/2	4	3/4	7/8	1 1/4	1 1/2	15/16	3/16	1 5/16	2 1/16	3 1/16	3 1/8	2 1/8	3 5/8	2	3.18	1 1/2
2	2 15/16	2 3/8	1 15/16	7	5	4	7/8	1	1 5/16	1 9/16	15/16	3/16	1 3/8	2 1/16	3 9/16	3 5/8	2 5/8	4	2 3/8	3.53	2
2 1/2	3 5/16	2 7/8	2 5/16	7 3/4	5 7/8	4	7/8	1	1 3/8	1 5/8	1 1/16	3/16	1 7/16	3 1/16	4 3/16	4 1/4	3 1/8	4 7/8	2 3/8	4.15	2 1/2
3	4 1/16	3 1/2	2 15/16	8 1/4	6 5/8	8	3/4	7/8	1 7/16	1 3/4	1 3/16	1/4	1 1/2	3 13/16	4 13/16	4 7/8	3 3/4	5 3/4	2	2.53	3
3 1/2	4 5/8	4	3 3/8	9	7 1/4	8	3/4	7/8	1 1/2	1 7/8	1 3/16	5/16	1 9/16	4 5/16	5 5/16	5 3/8	4 1/4	6 3/8	2	2.77	3 1/2
4	5 1/16	4 1/2	3 13/16	10	7 7/8	8	7/8	1	1 5/8	2	1 3/16	5/16	1 11/16	4 13/16	5 13/16	5 7/8	4 3/4	6 7/8	2 3/8	3.02	4
4 1/2	5 11/16	5	4 5/16	10 1/2	8 1/2	8	7/8	1	1 11/16	2 1/8	1 5/16	3/8	1 3/4	5 5/16	6 5/16	6 3/8	5 1/4	7 1/2	2 3/8	3.25	4 1/2
5	6 5/16	5 5/16	4 13/16	11 1/2	9 1/4	8	1	1 1/8	1 3/4	2 1/4	1 5/16	7/16	1 13/16	6 1/16	7 1/16	7 1/8	6	8 1/8	2 3/8	3.54	5
6	7 3/8	6 5/8	5 3/4	12 3/4	10 5/8	12	7/8	1	1 7/8	2 3/8	1 7/16	7/16	1 15/16	6 15/16	7 15/16	8	6 7/8	9 5/8	2 3/8	2.76	6
7	8 1/2	7 5/8	6 5/8	14	11 7/8	12	1	1 1/8	2	2 1/2	1 5/8	7/16	2 1/16	8 1/16	9 1/16	9 1/8	8	10 3/4	2 3/4	3.08	7
8	9 5/8	8 5/8	7 5/8	15 1/2	13	12	1 1/8	1 1/4	2 1/8	2 5/8	1 5/8	7/16	2 3/16	9 3/16	10 7/16	10 1/2	9 1/8	11 3/4	3	3.36	8
10	12 1/8	10 3/4	9 3/4	17 3/4	15 1/4	16	1 1/4	1 3/8	2 1/4	2 7/8	1 13/16	9/16	2 5/16	11 5/16	12 13/16	12 7/8	11 1/4	13 7/8	3 1/8	2.98	10
12	14 3/8	12 3/4	11 3/4	20 1/2	17 3/4	16	1 3/8	1 1/2	2 1/2	3 1/8	2	9/16	2 9/16	13 7/16	14 15/16	15	13 1/8	16 1/4	3 1/4	3.46	12
14	15 5/8	14	12 7/8	23	20 1/4	20	1 1/4	1 3/8	2 7/8	3 1/2	2 5/16	9/16	2 15/16	14 7/16	15 15/16	16	14 3/8	19 7/8	3 1/8	3.17	14
15	16 5/8	15	13 3/4	24 1/2	21 1/2	20	1 3/8	1 1/2	3	3 5/8	2 3/8	9/16	3 1/16	15 11/16	17 3/16	17 1/4	15 5/8	20	3 1/4	3.37	15
16	17 1/2	16	14 1/2	25 1/2	22 1/2	20	1 1/2	1 5/8	3 1/4	4	2 11/16	11/16	3 3/16	16 13/16	18 1/16	18 1/2	16 3/4	20 7/8	4 1/4	3.52	16
18	19 3/4	18	16 1/2	28	24 3/4	24	1 1/2	1 5/8	3 1/2	4 1/4	2 13/16	11/16	3 3/16	18 13/16	20 11/16	20 3/4	18 3/4	23 1/8	4 1/4	3.23	18
20	22	20	18 1/2	30 1/2	27	24	1 5/8	1 3/4	3 5/8	4 3/8	2 7/8	11/16	3 11/16	20 13/16	22 11/16	22 3/4	20 3/4	25 1/4	4 1/2	3.53	20
22	23 3/4	22	20	33	29 1/4	24	1 3/4	1 7/8	3 7/8	4 11/16	3 1/16	3/4	3 15/16	23 3/16	25 5/16	25 1/4	23 1/8	27 3/8	4 3/4	3.81	22
24	26	24	22	36	32	24	1 7/8	2	4 1/8	4 15/16	3 1/4	3/4	4 3/16	25 1/16	27 3/16	27 1/4	25	30	5	4.18	24

TABLE III. DIMENSION TABLE RECOMMENDED FOR AMERICAN DOUBLE EXTRA HEAVY 600-LB. PRESSURE STANDARD, TONGUE AND GROOVE JOINT

Pipe Size	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P	Q	R	S	T	U	Pipe Size
½	1 5/16	27/32	9/16	4 1/4	3	4	1/2	5/8	7/8	1 1/8	11/16	3/16	15/16	13/16	1 7/16	1 1/2	3/4	1 3/4	1 5/8	2.12	½
¾	1 1/2	1 1/16	¾	4 3/4	3 3/8	4	1/2	5/8	1	1 1/4	3/4	3/16	1 1/16	1 5/16	1 15/16	2	1 1/4	2 1/8	1 5/8	2.38	¾
1	1 27/32	1 5/16	31/32	5 1/2	4	4	5/8	3/4	1 1/8	1 3/8	7/8	3/16	1 1/16	1 1/2	2 1/4	2 5/16	1 7/16	2 5/8	1 3/4	2.83	1
1 1/4	2 5/32	1 21/32	1 3/32	6	4 3/8	4	5/8	3/4	1 1/4	1 1/2	15/16	3/16	1 1/16	1 13/16	2 11/16	2 3/4	1 3/4	3	1 3/4	3.09	1 1/4
1 1/2	2 1/2	1 23/32	1 1/2	6 3/4	4 7/8	4	3/4	7/8	1 3/8	1 11/16	1	1/4	1 1/16	2 1/16	3 1/16	3 1/8	2 1/8	3 5/8	2	3.44	1 1/2
2	3 1/16	2 3/8	1 15/16	7 1/2	5 7/8	8	5/8	3/4	1 1/2	1 13/16	1 3/16	1/4	1 1/16	2 1/16	3 1/16	3 5/8	2 5/8	4	1 3/4	2.25	2
2 1/2	3 1/16	2 7/8	2 5/16	8 1/2	6 1/2	8	3/4	7/8	1 3/4	2 1/16	1 5/16	1/4	1 13/16	3 3/16	4 3/16	4 1/4	3 1/8	4 7/8	2	2.48	2 1/2
3	4 5/16	3 1/2	2 15/16	9 1/2	7 1/2	8	7/8	1	1 7/8	2 1/4	1 1/2	5/16	1 15/16	3 13/16	4 13/16	4 7/8	3 3/4	5 3/4	2 3/8	2.87	3
3 1/2	4 7/8	4	3 3/8	10 3/4	8 1/2	8	7/8	1	2	2 3/8	1 1/2	5/16	2 1/16	4 5/16	5 5/16	5 3/8	4 1/4	6 3/8	2 3/8	3.25	3 1/2
4	5 5/16	4 1/2	3 13/16	11 1/2	9 1/4	8	1	1 1/8	2 1/4	2 5/8	1 13/16	5/16	2 5/16	4 13/16	5 13/16	5 7/8	4 3/4	6 7/8	2 3/4	3.54	4
4 1/2	5 15/16	5	4 5/16	13	10 1/2	8	1	1 1/8	2 3/8	2 7/8	1 13/16	7/16	2 7/16	5 5/16	6 5/16	6 3/8	5 1/4	7 1/2	2 3/4	4.01	4 1/2
5	6 1/16	5 5/16	4 13/16	14	11 1/2	8	1 1/8	1 1/4	2 1/2	3	2	7/16	2 1/16	6 1/16	7 1/16	7 1/8	6	8 1/8	3	4.40	5
6	7 5/8	6 5/8	5 3/4	15	12 1/2	12	1 1/8	1 1/4	2 3/4	3 1/4	2 1/4	7/16	2 13/16	6 15/16	7 1/16	8	6 7/8	9 5/8	3	3.23	6
7	8 5/8	7 5/8	6 5/8	16 1/2	13 3/4	12	1 1/8	1 1/4	2 7/8	3 3/8	2 1/4	7/16	2 15/16	8 1/16	9 1/16	9 1/8	8	10 3/4	3	3.56	7
8	9 7/8	8 5/8	7 5/8	18 1/2	15 1/2	12	1 1/4	1 3/8	3	3 1/2	2 5/16	7/16	3 1/16	9 3/16	10 7/16	10 1/2	9 1/8	11 3/4	3 1/8	4.01	8
10	12	10 3/4	9 1/2	21 1/2	18 1/2	16	1 1/4	1 3/8	3 3/8	3 7/8	2 11/16	7/16	3 7/16	11 7/16	12 7/16	12 1/2	11	13 7/8	3 1/8	3.60	10
12	14 3/8	12 3/4	11 1/2	24	21	20	1 1/4	1 3/8	3 1/2	4 1/8	2 13/16	9/16	3 7/16	12 13/16	14 3/16	14 1/4	12 3/4	16 1/4	3 1/8	3.28	12
14	15 5/8	14	12 1/2	25 1/4	22	20	1 3/8	1 1/2	3 5/8	4 1/4	2 15/16	9/16	3 11/16	14 1/16	15 1/16	15 5/8	14	19 7/8	3 1/4	3.44	14
15	16 3/4	15	13 1/2	27 1/4	23 3/4	20	1 3/8	1 1/2	3 7/8	4 1/2	3	9/16	3 15/16	15 1/16	16 1/16	16 5/8	15	20	3 1/4	3.71	15
16	17 3/8	16	14	27 3/4	24 1/4	20	1 1/2	1 5/8	4	4 3/4	3 3/16	11/16	4 1/16	15 15/16	17 1/16	17 1/2	15 7/8	20 7/8	4 1/4	3.79	16
18	19 1/2	18	16	31	27	20	1 5/8	1 3/4	4 1/4	5	3 1/4	11/16	4 3/16	17 15/16	19 11/16	19 3/4	17 7/8	23 1/4	4 1/2	4.22	18
20	21 3/4	20	18	33 3/4	29 1/2	20	1 3/4	1 7/8	4 1/2	5 1/4	3 7/16	1 1/16	4 7/16	19 15/16	21 13/16	21 7/8	19 7/8	25 1/4	4 3/4	4.61	20

as far as drilling and number of bolts are concerned, but that the flange diameter and thickness and size of bolts be calculated to withstand the various working pressures with a satisfactory factor of safety. This standard for flanges should be known as the American extra heavy standard as heretofore, except that the pressures be designated in connection with it.

It is further recommended that the present 1200-lb. hydraulic standard be adopted for pressures ranging above 500 lb. up to 1200 lb., inclusive, as far as drilling, wide diameter and number of bolts is concerned, and that the flange diameter and thickness and size of bolts be calculated to provide for a satisfactory factor of safety for the various pressures under consideration. This standard should be known as the American double extra heavy standard, as heretofore, except that the pressures be designated with it.

These recommendations would provide but two standards from 250 lb. to and including 1200 lb., inclusive, in steam service. This allows of due consideration for the specifying engineer, the operator and the manufacturer.

Shouldered stud bolts, as shown in the cut, need not be specified for pressures below 400 lb. If bolts are made from the higher grades of steel now on the market, having considerably more, and in some cases twice the elastic limit and strength of the ordinary hot drawn bar material, on the basis of which the recommendations given herewith for bolting have been worked out, the factor of safety of 12 will be greatly increased.

These recommendations for high pressure standards for flanges and bolting take into consideration the important factor of locating the bolt circle as close to the fitting or pipe wall as possible, which tends to eliminate to a great extent the unequal temperature between the bolting

and the piping proper. Furthermore, this would minimize the unequal stress or distortion of the flanges and bolting at times of heating and cooling in a piping system. At the same time the accessibility and the necessary fillets in castings between flanges and fittings or valve necks must be taken into consideration, not only from the standpoint of strength and safety but from a manufacturing and construction standpoint as well. The bolt circle must be of sufficient diameter to facilitate the erection of the valve or fitting without the use of a special wrench and should permit the use of a standard nut on the bolt without machining away the entire fillet on the casting.

Depreciation of Equipment

By James Gibbons

IN SPITE of the so-called rules for figuring depreciation and estimating values, there is no scientific standard by which the validity of these more or less arbitrary rules can be checked. Any one who attempts to make a valuation without taking into full consideration the purpose for which the valuation is made, is ignoring the very important fact that all valuations are relative and not absolute. That is, the value of a piece of machinery is dependent not only upon its first cost and physical state, but upon its utility and desirability, under certain assumed conditions. For instance, a new machine costing $1000, which had been dumped into the middle of the Sahara Desert, might be reasonably valued at zero, while delivered in a machine shop, where it was needed, it would be worth at least $1000. Roughly speaking, there are two ways of answering the question as to what a thing is worth. The first is, that it is worth what it will bring, but this depends to a large degree upon the conditions under which it is offered for sale, and as for accounting purposes a sale is, as a rule, not contemplated. When we are dealing with a plant, the conditions of a hypothetical sale are to a large extent subject to individual choice.

Reasoning from a second angle, the real value of a piece of equipment is its investment value, depending to a considerable degree upon its actual or potential utility as a dividend payer, with a limiting value made up of the sum of all money spent on the machine installed in operating condition, less a proper depreciation to cover the cost of replacing the machine when it is worn out.

This method is, in the opinion of the writer, the more correct for accounting purposes, and as the erection cost is a capital expenditure which is expected to pay dividends, there would appear to be no reason for treating it differently from any other asset, except that it may be assumed to suffer the same depreciation as the machine itself.

This is based upon the assumption that the machine can be profitably used, for it was purchased with this end in view, and it would appear reasonable to make the standard accounting practice conform to it. If the machine should prove to be not profitable, it would seem to be more reasonable to charge the cost off as a loss of assets due to an error in judgment, rather than to charge it to the machine, and indirectly to the cost of production.

Leaking Steam Line Repaired by Welding

WHEN THE Ladew Leather Co., Newark, N. J., decided to replace a section of 8-in. high pressure steam line, which was leaking in the threads at a flange, they planned on doing the work by the old methods.

FIG. 1. SECTION OF 8-IN. PIPE CUT OUT OF STEAM LINE.

At the opportune time, however, an oxy-acetylene engineer happened to call on them. Learning of the leaky pipe, and the company's proposed methods of repair, he suggested cutting out the defective pipe and welding in a new section by the oxy-acetylene process. He showed them how much time and money could be saved by doing the job this way. As they were a trifle incredulous at first about doing the work themselves, he offered to supervise the job and instruct their operator.

First, the steam was shut off and all drips were opened. Then using an Oxweld cutting blowpipe, a section of pipe which included two flanges was cut out

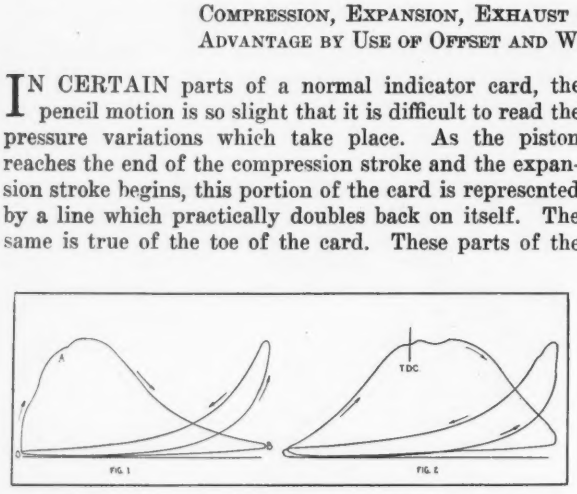
FIG. 2. THE COMPLETED WELD, SHOWING NEW SECTION OF PIPE IN PLACE

about 8 in. back of each flange. The cut edges were beveled and the new section was welded into place, the engineer instructing the company's operator in the proper methods of pipe welding. The entire job was completed in 1 hr. and 15 min. at a cost of not more than $5, and according to the company's superintendent saved them at least $100 on the repair alone, besides a several hour shutdown.

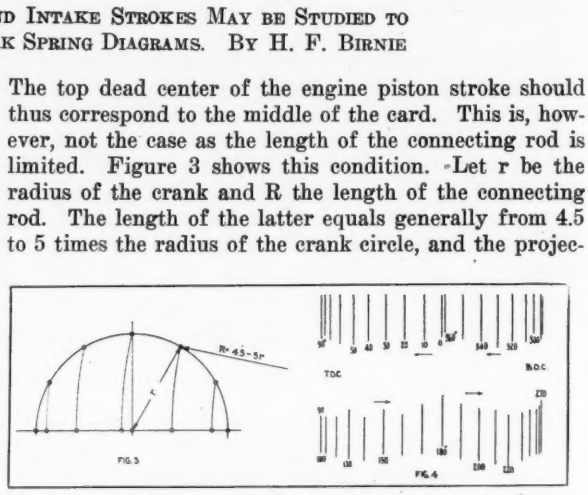
Oil Engine Operation Studied by Special Cards

COMPRESSION, EXPANSION, EXHAUST AND INTAKE STROKES MAY BE STUDIED TO ADVANTAGE BY USE OF OFFSET AND WEAK SPRING DIAGRAMS. BY H. F. BIRNIE

IN CERTAIN parts of a normal indicator card, the pencil motion is so slight that it is difficult to read the pressure variations which take place. As the piston reaches the end of the compression stroke and the expansion stroke begins, this portion of the card is represented by a line which practically doubles back on itself. The same is true of the toe of the card. These parts of the diagram are particularly important in the study of Diesel engine characteristics, and so cards of a special type are used.



FIGS. 1 AND 2. TWO OFFSET CARDS COMPARED WITH NORMAL CARDS



FIGS. 3 AND 4. METHOD OF DETERMINING THE TOP DEAD CENTER OF AN OFFSET CARD

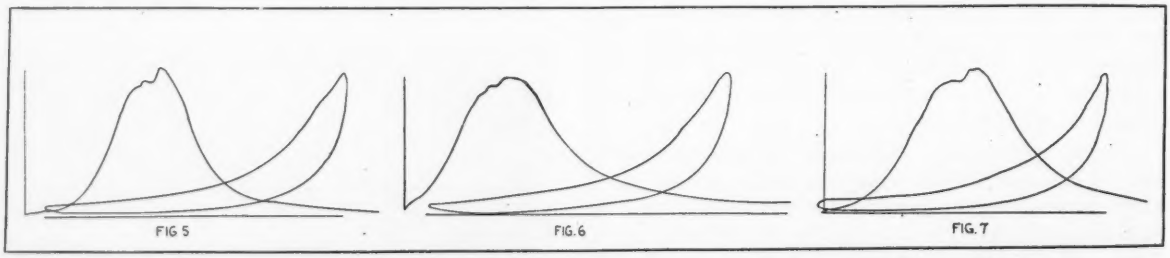
In order to permit a detailed study of these points on the card, it is necessary to take what is sometimes called an offset card. In taking such a card, the indicator motion must be taken from a point other than the piston or crosshead motion. One method is use a drive from one of the other cylinders which will give the maximum drum motion at the end of compression and the beginning of the power stroke so that the diagram will be stretched out at this point. Obviously, however, such cards cannot be used to compute the indicated horsepower of the engine.

Figure 1, for instance, shows a normal and an offset diagram taken on the same card of a three-cylinder engine running at three-quarters load. The normal diagram was of course taken with the proper cam and for the offset diagram, the indicator drum was driven by the cam of one of the other cylinders.

OA represents the compression-stroke, A being the end of the compression. The moment where the fuel is injected is shown clearly and the raise of the offset-curve after A indicates either a lead that is a little too big or an injection air pressure somewhat too high for obtaining a perfect Diesel-cycle.

Figure 2 shows an offset-card taken with an auxiliary indicator driving-cam set at 90 deg. to the normal cam. The top dead center of the engine piston stroke should thus correspond to the middle of the card. This is, however, not the case as the length of the connecting rod is limited. Figure 3 shows this condition. Let r be the radius of the crank and R the length of the connecting rod. The length of the latter equals generally from 4.5 to 5 times the radius of the crank circle, and the projections of the points on one side of the dead center will not be the same as those on the other side.

One way to find the point on the offset card corresponding to this top dead center, is to rig up the indicator to the 90 deg. offset cam, bar over the engine, and mark on the diagram points corresponding to a certain travel of the engine piston. Figure 4 shows such a chart where points equivalent to 10 deg. were marked on the flywheel. The length of the offset diagram will not always correspond to the length of the chart; when the engine is running, the mass acceleration of all the moving parts of the indicator rigging, such as drum, transmission rolls, etc., are liable to elongate it.



FIGS. 5 TO 7. OFFSET AND NORMAL CARDS FOR EACH CYLINDER OF A THREE-CYLINDER ENGINE

Another and more practical way to take an offset diagram together with the normal indicator card, is the following:

Take the normal diagram but leave the indicator cock open.

Disconnect the indicator from the indicator string.

Hold the indicator pencil against the card and pull the indicator drum by hand.

With a little practice one will be able to pull with the right speed so as to get the offset diagram near the middle of the card.

Figures 5, 6 and 7 were taken on a three-cylinder engine. The indicator was driven by a string from three different cams on the vertical shaft. The power strokes on the normal cards seem rather different, pointing to a

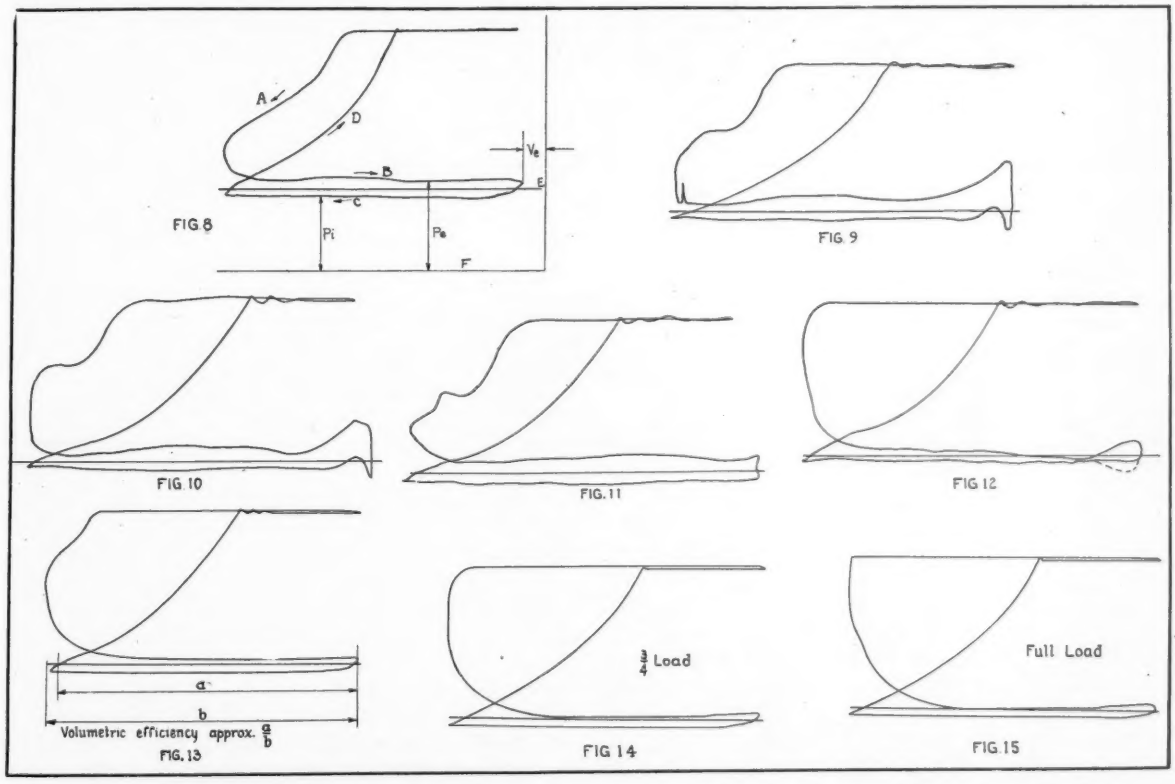
pronounced after-burning in Fig. 6. The pull cards, however, show that the power-stroke was practically the same in all the cylinders.

Intake and Exhaust Stroke Studied with Weak Spring Diagram

Another important part of a card is that represented by the exhaust and air intake stroke. Since the pressure is low during these strokes, the ordinary indicator spring is too stiff to allow the variations in pressure to show on a scale large enough to read. Normal diagrams are made with heavy springs such as 300, 400 or 500-lb. springs and it is obvious that variations in pressure of the order of 3 or 4 lb. will not show.

The surface of the diagram during the exhaust and intake strokes represents negative work, and the smaller this surface is kept, the bigger will be the work done by the engine.

Figure 8 shows an ordinary weak spring diagram, where A is the end of the expansion stroke, B the exhaust stroke, C the air intake stroke, D the beginning of the compression stroke, E the line of atmospheric pressure and F the zero pressure line, assuming the atmospheric pressure to be normal, in other words 14.7 lb., per square inch. At the end of the exhaust stroke, a certain volume V_e of the exhaust gases is remaining in the cylinder, under the pressure P_e. Before the intake air is allowed to rush into the cylinder, these exhaust gases must ex-



FIGS. 8 TO 15. WEAK SPRING DIAGRAMS PERMIT A DETAILED STUDY OF THE LOW PRESSURE PART OF THE CARD

In order to overcome this difficulty, weak spring diagrams may be taken. As a matter of fact, however, it is better to take such diagrams with a fairly heavy spring, but with a big indicator piston. This increases the travel of the pencil so that 1 in. of travel represents a smaller pressure than in the normal card. The advantage of this method lies in the fact that the heavy spring will counteract the vibrations of the pencil motion which might be encountered if a weak spring were used. Most indicators are furnished with several sizes of pistons and cylinders which are easily changed. There should be, of course, a stop limiting the upward travel of the piston.

If one seeks to get the indicated horsepower of an engine, so as to get either its mean indicated pressure, its mechanical or thermal efficiency, one should consider the work given to the engine by the flywheel and subtract it from the work produced by the power-stroke. pand to the pressure P_1, thereby reducing the volume of fresh air getting into the cylinder.

The volume of exhaust gases and fresh air filling the cylinder at the end of the air intake stroke is constant, but the weight of this air will vary with the intake temperature and intake pressure. It follows, thus, that with a low P_e and a high P_1 the load capacity of a cylinder will be increased. A low intake pressure P_1 will also decrease the compression height in the cylinder and an early closing of the exhaust valve will raise the pressure P_e.

Figures 9, 10 and 11 illustrate a bad case. The pressure at the end of the exhaust stroke was so high that the intake valve had to open against 10 lb. pressure and made a sharp snapping noise. By closing the exhaust valve later, the noise was eliminated; but the surface of negative work remained large as, no broader

cam being available, the high pressure at the end of the exhaust stroke had to be corrected by opening the exhaust valve later.

Figure 12 also shows an exhaust valve closing too early but the general conditions are much better than in Fig. 11. A late opening of the intake valve would be shown by the dotted line.

Near the end of the expansion stroke, the exhaust valve begins to open and the gases rush out. Their speed is independent of the size of the exhaust valve opening. The square of this speed is proportional to the amount of heat contained in the exhaust gases and inversely proportional to their weight. The weight remains about constant, but the heat will increase with the load, in other words, the speed of the exhausting gases will be higher at full than at light load.

This is why the mean pressure p_e of the exhaust stroke is higher at light load than at full load. Figures 12, 13 and 14 illustrate this fact, though the pressure at the end of the exhaust stroke is slightly higher at full load than at light load. This may be explained by the fact that the cylinder being hotter at full load, the pressure of the gases remaining in the cylinder is somewhat increased.

Figures 16, 17, 18 and 19 represent four weak spring diagrams taken at quarter, half, three-quarter and full load. AB is the exhaust stroke, BC the intake stroke and AA_1 the line showing the upward travel limit of the indicator plunger. The horizontal line and the middle of ABC is the atmospheric pressure line.

It is probable that the wavy beginning and steep end line of the exhaust stroke are due to the mass acceleration of the indicator's mechanism. This wave is more pronounced at heavy load. The irregular tracing of the exhaust line can also be originated by vibrations of the exhaust gas column. A simple indicator, though, is not apt to settle the question. Such vibrations are to be measured with special devices and are generally produced in long exhaust pipes without bends.

The raise in pressure at the end of the exhaust stroke is, in the four previous figures, not caused by the exhaust of the other cylinders, as they were taken on a three cylinder engine and the exhaust valve closing of one cylinder did not overlap with the opening of the next.

Figure 20 shows a full load card taken on the same engine but with the engine slowed down to half of the r.p.m. The waves of the expansion stroke are still more exaggerated as the indicator drum revolves so much slower; but as the exhaust valve kept open twice as long, the end pressure of the exhaust is considerably lower.

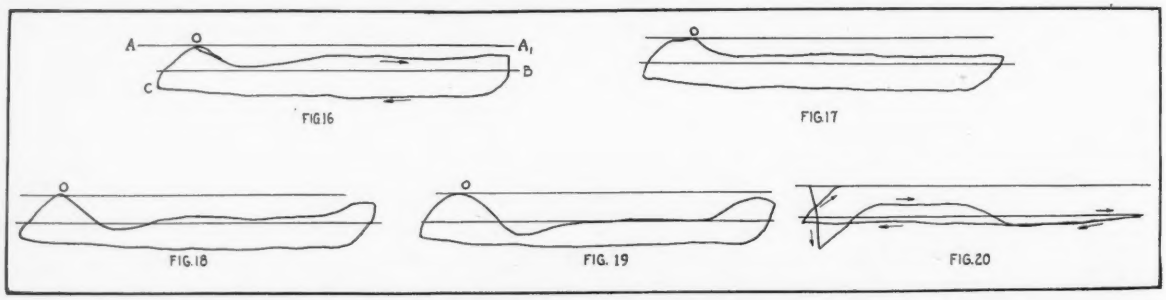


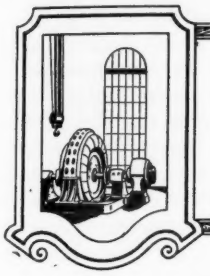
FIG. 16 TO 20. VARIOUS LOAD CONDITIONS SHOWN WITH WEAK SPRING CARDS

Founders' Trade Practice

ENGINEERS who frequently find it necessary to have castings made, will be interested in the following standard trade practices adopted by the five companies forming the Electric Steel Founders' Research Group at their recent meeting in Pittsburgh, Pa., on June 16:

1. Suitable pattern equipment for economical molding shall be furnished by customer.
2. The foundry will not be responsible for correctness of pattern equipment to blueprint, except when such equipment is made for the customer under the supervision of the foundry.
3. Repairs on pattern equipment shall be paid for by the customer, except when occasioned by carelessness on the part of the foundry.
4. Patterns must have distinctive colors to identify separately the core-prints, machined surfaces, and rough casting.
5. All patterns, coreboxes, and loose pieces thereof, must be properly numbered for identification.
6. All transportation charges on pattern equipment to and from foundry, shall be paid by the customer.
7. The foundry will not carry insurance on customers' pattern equipment.
8. Free replacement will be made of defective castings if reported and returned to foundry within a reasonable time.
9. The foundry will not be responsible for any expense on defective castings, incurred by the customer.
10. The customer will be charged with the cost of molds and cores discarded by the foundry, due to change in patterns or coreboxes.

USE OF NON-CONDENSING turbine units or bleeder type turbines in conjunction with heating systems, either central station or industrial, has received considerable general attention in the past two years. In some cases a non-condensing turbine has been installed as a reducing valve to lower the steam pressure to a value usable in the heating mains. The usual custom is to control, either by hand or automatically, the amount of the electrical energy generated so that just sufficient low pressure steam is produced to meet heating requirements. If furnished with automatic control, the usual design includes an arrangement for the pressure in the exhaust main to act upon a diaphragm which in turn controls a pilot valve controlling the opening of the steam throttle valve.



ELECTRICAL MACHINERY FOR LIGHT AND POWER

Remotorizing Plant Raises Power Factor

REMOTORIZING EQUIPMENT IN THIS STEEL PLANT RESULTED IN RAISING THE POWER FACTOR FROM AROUND 30 TO OVER 60 PER CENT. BY O. H. HENSCHEL

IN THE SELECTION of electric motors for industrial use the factor of providing the requisite horsepower is not the only one to be taken into consideration. Employing machines of proper characteristics is fully as important if proper operation, low maintenance costs and minimum energy costs are to be obtained and maintained.

Up to the present time, even among engineers, it has been more or less customary to apply some form of rule of thumb or even in many instances, as was the case in one plant, pure guess work, when selecting motors, and as a consequence we today find many drives on which the motors are entirely too large for the work to be done and frequently of wrong characteristics for the particular kind of service to be rendered. The result is low efficiency accompanied by useless consumption of electric energy, and in the case of alternating current motors, low power factor with its detrimental effect upon the whole system.

An example of improper motoring was, up to a few months ago, in evidence in a mid-west manufacturing plant devoted to the making of heavy punched, pressed and drawn steel products primarily such as automobile brake drums, hub flanges, step hangers, etc., etc. Essentially this plant consists of the following departments: press shop, annealing and pickling, welding, tool room and machine shop. In all, approximately 60 motors are in use, ranging from ¾ to 250 hp. They are all served by 3-phase, 60-cycle energy delivered to the plant over the lines of the local public service company at 13,200 v. but stepped down to 440 v. through the medium of a 400 kv.a. oil-insulated air-cooled transformer.

In addition to the motors there is also a 100-kw., 6-phase unity-power factor rotary converter which at first was utilized only to charge the batteries of the company's industrial trucks and provide direct current for the operation of a 3-T. traveling hoist equipped with an electro magnet for handling steel stock. Even with all of the batteries on charge and the hoist working, the maximum load on the converter was not in excess of 20 per cent of its rated capacity.

DISCOVERY OF TROUBLE

Evidently little had ever been thought of and less done in reference to providing the proper sizes and types of motors in this plant and undoubtedly little would ever have been done had not the local utility company installed a reactive watthour meter, discovered an unbelievably low power factor and penalized the company. Just where the heart of the trouble existed was at first hard to determine, but investigation soon disclosed the fact that both power and hydraulically operated presses in the press shop were over-motored, in some instances as much as from 300 to 400 per cent. In other words, all of the motors were running on an average of but about 20 per cent of their rated capacity and corresponding degree of efficiency; the cause of the low system power factor was, of course, then at once quite obvious.

POWER REQUIREMENT DATA, SHOWING CHANGES MADE AND THEIR EFFECT UPON POWER FACTOR

Machine Driven	Original Motor					Motor Installed				
	Type	Hp.	Hp. Required	% P.F.	% Eff.	Type	Hp.	% P.F.	% Eff.	Average % Gain P.F.
Bliss No. 72½ 800 Tons 10 in. Stroke						High Torque	6.6	84.6	88	
Bliss No. 72½ 800 Tons 10 in. Stroke	Squirrel Cage Induction	25	1.Load 5.6 F.Load 8.9	44 44	88	High Torque	6.6	88	90	44
Bliss No. 72½ 800 Tons 10 in. Stroke	Squirrel Cage Induction	20	1.Load 3.3 F.Load 4.7	48 89	85	High Torque	6.6	88	90	43
Bliss No. 72½ 800 Tons 10½ in.Stroke	Squirrel Cage Induction	15	1.Load 2.77 F.Load 3.69	84 88	84	S-High Torque	13	88	90	48
Bliss No. 77½ 800 Tons 12 in. Stroke	Squirrel Cage Induction	20	1.Load 2.96 F.Load 3.18	88 88	88	High Torque	6.6	89	90	60
Bliss No. 77½ 800 Tons 12 in. Stroke	Squirrel Cage Induction	20	1.Load 2.43 F.Load 3.80	87 40	88	High Torque	6.6	89	90	61
Bliss No. 78½ 88 Tons 8 in. Stroke	Squirrel Cage Induction	5	1.Load 0.93 F.Load 4.07	12 84	88	High Torque	6.6	89	90	78
Bliss No. 80½ 1000 Tons 20 in. Stroke	Slip Ring Induction	75	1.Load 12.9 F.Load 46.8	33 75	87	Slip Ring Induction	80.0	84	89	6
Bliss No. 27 1000 Tons 8½ in. Stroke	Squirrel Cage Induction	25	1.Load 1.62 F.Load 3.84	14 82	89	Not as yet remotored				
Toledo No. 644 1000 Tons 8½ in. Stroke	Squirrel Cage Induction	25	1.Load 3.26 F.Load 7.67	11 84	87	Not as yet remotored				
Elmes High Press.Pump	Slip Ring Induction	250	1.Load 52.8 F.Load 208.5	20 87	91	Now fitted with 150 hp. motor. No tests as yet made.				
Line shaft	Squirrel Cage Induction	25	1.Load 6.3 F.Load 14.0	47 74	88	Now fitted with 17½ hp. D.C. motor				
Air Comp. 10 x 12 in. Single Cyl.	Slip Ring Induction	41	1.Load 19.9 F.Load 41.7	87 84	89	Now fitted with 35 hp. D.C. motor Press. reduced from 110 to 75 lb.				
Split Shear	Squirrel Cage Induction	15	1.Load 1.4 F.Load 3.6	15 72	88	Now fitted with 7½ hp D.C. motor				

Tests were conducted on all of the motors under consideration, these tests being made with the machines working under normal load and in some instances under overload conditions. The results of these tests are shown herewith and, as may be seen in some few instances, motors had been used, especially on the presses, of a size as much as seven times larger than required. An example of this was the case of one Bliss No. 27 straight-side toggle-joint power press having a tonnage of 1000, a stroke of 2½ in. and making from 500 to 600 impressions per hour. This press is employed exclusively for blanking and, naturally, under the most severe conditions the load is comparatively light.

One of the largest power presses installed at this plant is a straight-side Bliss No. 80½ having a tonnage

of 1000, a stroke of 20 in. and an actual production of 275 pieces per hr. Under the heaviest load obtainable at the time, namely, the forming of tractor rocker beams, the maximum horsepower required was 49.8; the driving motor, a slip ring machine, was rated at 75 hp. This has now been replaced with a 50-hp. motor capable of developing 61 hp. for short periods of time.

Hydraulic pump equipment consists essentially of three Elmes units, each of the duplex four-plunger type, connected to driving motors through the medium of a herringhone gear and pinion. These pumps, which operate at about 100 strokes per min., are capable of maintaining pressures of from 400 to 500 lb. per sq. in. and except for one which has been driven by a 150-hp. motor, were each connected to 250-hp. motors. All of these machines were of the variable speed slip-ring type.

Due to the nature of the product manufactured and, therefore, because of the variation in load, it was quite

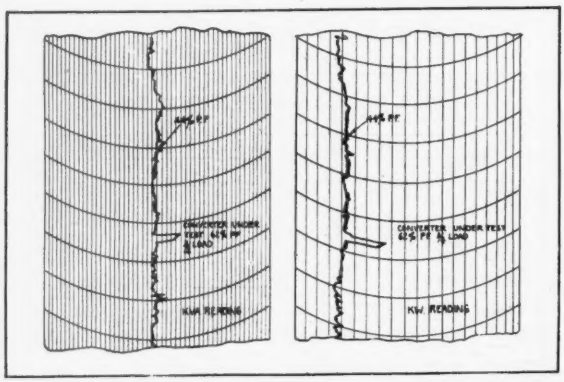


FIG. 1. SHOWING EFFECT OF LOADING ROTARY CONVERTER UPON POWER FACTOR

difficult to determine just what the requirements of these units might be. However, tests which were conducted with what was considered heavy work in the press indicated a maximum of approximately 115 hp.

In the accompanying table are indicated the various machines installed, the original size and type of motor utilized and the performance of each, such as horsepower required to operate, the power factor and assumed efficiency. It also indicates at the right the size and type of motor used to replace the original machine.

Peculiarity of Service

In the operation of punch presses, particularly when doing light work, the kinetic energy accumulated and stored in the flywheel is usually more depended upon to complete the actual operation than is the power delivered by the motor, although it is true that with some classes of heavy work, especially in the case of long draws, the motor is to some extent required to help out. It was found, however, in the case of Bliss Nos. 77½ and 78½ straight-side presses, particularly when used for blanking and light draws, that the actual power required during the period of operation was but little in excess of that required to operate the press empty but with the clutch in; that is, with the machine in gear.

After having determined the number of horsepower necessary to operate these Bliss Nos. 77½ and 78½ presses, a trial motor of a commercial type commensurate with the requirements as indicated by the tests conducted, was installed. In the case of one of these machines, a Bliss No. 77½ driven by a 20-hp. squirrel-cage induction motor, it was found from tests conducted that with the machine running out of gear the motor was required to deliver 2.36 hp. and with the machine working 3.18 hp. Accordingly, a 5-hp. squirrel-cage induction motor was installed, which, however, when tried failed to start the press; 7½-hp. slip-ring and squirrel-cage motors were then tried but the results were the same. It was thus evident that what was really required for these presses was a motor of low horse-power rating, but one having a high degree of power factor and an exceedingly high starting torque with about 15 per cent slip, a combination which for this particular kind of job was impossible to obtain in the open market. As a result, the problem was placed before numerous manufacturers of this kind of equipment, but of the many approached only two responded and while one of these did not obtain the contract for the building of these special machines because of a price difference in justice it must be admitted that the trial motor this manufacturer provided fulfilled the specifications fully as well as did the company whose motors were finally accepted.

Undoubtedly the problem might have been more easily solved by using an ordinary squirrel-cage induction motor of a size sufficient to give the required starting torque, but such procedure was precluded by the local electrical ordinance and the rulings of the public utility company, which prohibit the throwing of a motor of 7½ hp. and more across the service lines, and in addition to this the fact that no squirrel-cage induction motor of 7½ hp. would provide the necessary starting torque. Using a compensator naturally resulted in reduced terminal voltage, low initial current flow and consequently reduced torque, which in turn necessitated the use of a larger motor to give the necessary torque through a compensator. The consequence of this would have been the use of a motor much larger than required for operation and virtually a defeat of the object sought.

It is therefore obvious why it was necessary in preparing the specifications of these special motors to insist upon a horsepower rating of less than 7½. In fact, the horsepower of these machines had to be determined on a torque rather than a heat basis.

The Machines Selected

Motors installed on the various Bliss Nos. 77½ and 78½ presses and also on some of a few smaller presses of a size closely corresponding to a Bliss No. 75½ which had a tonnage of 200 and a 12-in. stroke, were designed and built to deliver 6½ hp. with a temperature rise of 50 deg. C. for 24 hr., a synchronous speed of 1800 r.p.m. and a full-load speed of 1600 r.p.m.; the full-load efficiency was 75 per cent, while the power factor from test was 73.4 per cent at one-half load, 83.4 per cent at three-fourths load, 88 per cent at full load, and 90.2 per cent at 25 per cent overload.

Practically indestructible silver-soldered rotors are employed while stator windings are of a design and construction as to withstand five times full-load current for 1 min. without injury.

Another unusual and perhaps radical scheme employed in connection with the installation of these motors is the comparatively short centers between the motors and press pulleys, which in each case was made 65 in.

The pulley ratio employed is about 1 to 5. To provide the desirable arc of contact on the motor pulleys, each machine has been fitted with an idler of the bracket-and-spring type attached directly to the motor frame. The writer has frequently been ridiculed for employing this short drive, even by men having many more years of power press experience than he, but up to the present time and after a period of over a year's experience, has found no cause for regretting having utilized such a scheme.

Each motor, together with the control, is mounted upon a concrete pedestal about 3 ft. high, an arrangement which, together with the shortened pulley centers, has resulted in the cutting down of the floor space occupied by the belts, motors and controls to one-third of that required. In addition to this the belt investment has been reduced in some instances as much as nine-tenths.

Costs

Comparison of initial costs of the new requirements and those replaced proves interesting. The first costs of the new motors were as follows:

Motor and control	$177.50
Belt	13.60
Installation	100.00
Total	$291.10

For even a 7½-hp, 900-r.p.m., high-torque machine with compensator, the costs would have been as follows:

Motor and control	$435.00
Belt	35.00
Installation	100.00
Total	$570.00

No comparison is made of wiring costs, this because of the fact that the wiring installed is in each instance practically heavy enough to care for a 25-hp. squirrel-cage induction motor. This was done to prevent an excessive drop in voltage which, particularly at moment of starting, is to be avoided in an installation of this kind if the maximum degree of torque is to be obtained. With a larger motor than would really be required, the difference in initial costs would be correspondingly greater.

Comparison of operating costs would be quite difficult to ascertain, but it has been shown that considering the plant as a whole the higher efficiency of these machines with their increased power factor has had a decided effect in raising the power factor of the system, reducing the demand and lowering the cost per unit of energy.

Limitations

It should not in general be inferred from this article that the writer would specify the use of this size of motor in connection with the presses mentioned for all kinds of work, for undoubtedly work could be placed in these presses which would tax the capacity of the motors, if not stall them entirely. The power required was unobtainable by mathematical means either for the work to be done or for that required to meet the maximum capacity of the presses. Men with many years of press experience were unable or dared not even venture a guess. Tests extending over a year's period were then conducted under all conditions of load and kind of jobs obtainable in this particular plant and it was upon the results of these tests and only upon the results of these tests that the actual power requirements were arrived at.

Should at any time work be done which would require power in excess of the capacity of the installed motors, a portable set consisting of a 20-hp. motor, switch and cut-out cabinet and compensator mounted on a skid can be placed back of the press desired and temporarily utilized for such particular jobs. This scheme is many times more economical and efficient than having a permanently installed motor carrying but 20—25 per cent load with correspondingly low efficiency and poor power factor.

Advantages of Smaller Motors

In general, the advantages accruing from the use of these smaller motors may be enumerated as follows:

(a) Improved power factor.
(b) Higher efficiency.
(c) Reduced energy consumption.
(d) Elimination of expensive compensators.
(e) Removal of actual motor control from hands of press operators.

Items (d) and (e) are of vital importance. In plants such as this where more or less inexperienced and much transient labor is employed, labor practically totally ignorant of the handling of electrical equipment, compensators, and therefore motors, also, are subjected to much abuse and consequent damage and frequent repairs. It is therefore desirable to take the actual starting of the motors out of the hands of the press operators as much as possible, this either by the entire elimination of compensators where the size of the motor will permit doing so or by the employment of automatic remotely controlled compensators.

Starters installed in connection with the 6½-hp. motors are of the magnetic type controlled from a push button located at the front left hand side of the press and are of the three-pole type equipped with no-voltage release and inverse-time-element overload relays with an adjustable calibration feature providing protection against overload on the motors.

Double Drive for Heavy Work

While the two Bliss No. 78½ presses referred to above have a stroke of 10 in., there is installed in this plant another, or third press of like make and manufacturer's number, but having a stroke of 16 in. Much heavy work, particularly long draw jobs, are frequently placed in this machine with the consequence that the 6½-hp. motors are too small for such type of work. In order, therefore, to avoid reverting to the old scheme of drive and compensator control, it is planned to utilize a double motor installation, each of which machines are of the type and horsepower rating of the others. An 8-in. belt will be used and a 9-in. pulley fitted to the motor shafts which are to be butted up against one another. Individual control boxes are to be installed, although these controls will be operated by a single "start" and "stop" button.

Hydraulic Pump Motors

As stated above, the required capacity of the hydraulic pump motors was obtained by placing the heaviest jobs securable in the presses, noting the maximum power consumption and selecting the next largest commercial

size of machine. This does, however, not give the size of motor required to operate the pump with the press fully loaded. In order to secure this information it would require blocking the press, allowing the pump to accumulate the maximum pressure permissible and observing the power intake from which, of course, knowing the efficiency of the motor, the horsepower output can be readily calculated. The result thus obtained would however, be the maximum and ought not to be the value of the horsepower rating of the motor to be used, this because of the fact that the maximum load is at any time but momentary. It is obvious, therefore, that the motor can have a horsepower rating considerably less than it might appear necessary. Assuming that with the pump working against a maximum pressure on the press, wattmeter readings would indicate a horsepower requirement of approximately 150, it would be unnecessary to use a motor of such rating. Particularly for short periods of actual working time such as encountered in heavy hydraulic press work, motors can well carry overloads of 50 per cent and more, so that in such instances as this a 100-hp. motor could well be used.

At best, unless systems of accumulators are employed, the loads on hydraulic press motors are very erratic, being about 10 to 15 per cent of the rated capacity of the machine when running idle and perhaps 150 to 175 per cent rated capacity at moment of operation. The actual load on such motors is maintained only about 10 per cent of the time; during the balance the machine is running idle. The effect of such operation on the efficiency and power factor is quite obvious.

Double-Cylinder Four-Plunger Pumps

Each of the pumps used at this plant for the operation of the hydraulic presses is of the double-cylinder, four-plunger, outside-packed type with plungers each 3 in. by 14 in. long and having a stroke of 12 in. One of these can serve either a 400-T., 8 ft. 2 in. stroke, four-column press or a 150-T., 41½-in. stroke, four-column press, and has been provided with a 150-hp., 580-r.p.m., slip-ring motor which replaced a 250-hp. unit of like type. Second and third pumps have now 100-hp. motors instead of 150 and 250-hp. motors heretofore used. One of these pumps serves a 150-T. press having a stroke of 5 ft. 9 in., while the other is used in connection with three presses, one of which has a rating of 150 T. and a stroke of 41½ in., while the other two are of like tonnage but with 42-in. strokes each.

Utilizing the Converter

As already stated, the rotary converter was carrying but a small load, so that in spite of carrying an overexcited field to establish a leading line current, the full effect of its power factor corrective possibilities was not utilized. To determine just what could be expected of this unit by adding load, a test was conducted by connecting a water rheostat on the direct current side of the converter and the taking of the kilovolt ampere and kilowatt graphic meter readings on the system service. The results of these tests indicated that with about 25 per cent load on the converter the system power factor was 38 per cent; with 75 per cent load, 48 per cent; with 100 per cent load, 53 per cent, and with 125 per cent load 62 per cent.

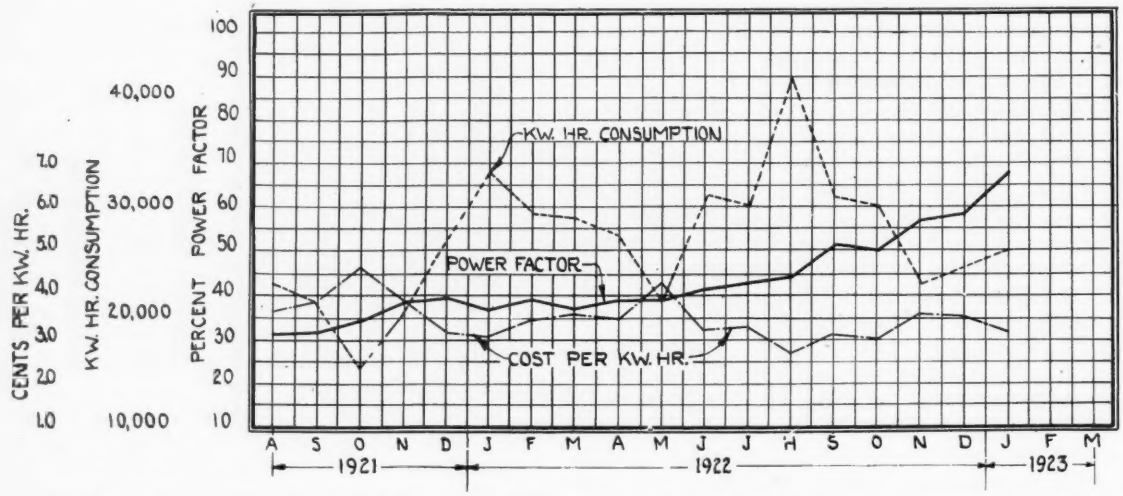


FIG. 2. ELECTRIC SERVICE COST DATA, SHOWING POWER FACTOR INCREASE AND ENERGY COST DECREASE

From the graphs shown in Fig. 1, it is apparent what effect the loading of the converter had upon the system power factor. At the time of taking readings the system power factor averaged about 44 per cent, while, as indicated, as soon as three-quarters load was placed on the machine the power factor at once increased to 62 per cent.

Local conditions, however, precluded the adding of much additional permanent load on the converter, although it was possible conveniently and economically to increase the load by the connection of an extra 50 hp., so that at present the average load on the machine is at times nearly 40 per cent of its rated capacity. This arrangement has undoubtedly assisted materially in helping to boost the system power factor. Just how much, however, is problematical and difficult of determination.

In general, it is estimated that with completion of the remotorization program now in progress, particularly on account of the variable nature of the load, the ultimate power factor will range from approximately 65 to 75 per cent. In order, therefore, to eliminate the

present penalization and reduce power costs to a minimum, it will eventually become necessary to install a corrective device either in the form of a synchronous motor or a static condenser.

Benefits to Be Derived

As is usually the practice, this company is charged not only a sliding rate for the actual kilowatt hour consumption of electric energy, but also a demand charge of $2.00 for the first 200 kw. and $1.50 for all in excess of 200 kw. In addition, a ruling of the public utility provides that "the maximum demand upon which monthly charge for power is based, hereinafter called the 'billed demand,' shall not be less than 75 per cent of the highest billed demand ascertained during the period of 12 mo. ended with the current month." It is thus obvious that when the actual demand drops materially the power costs will not vary according to the consumption. In fact, it has frequently occurred that under such conditions the unit power costs increased rather than decreased.

As the billed demand is arrived at by dividing the product of the metered demand and 80 (standard power factor) by the actual power factor, any increase in actual power factor will tend to reduce the billed demand and, consequently, the demand charge, and with it the unit energy cost. In any event, in spite of increased power factor, a marked variation of electric energy consumption during any single year is bound to raise havoc with the power bill.

Proof of the savings to be secured by improving the power factor may best be realized by citing a specific example.

During the month of January, 1922, the kilowatt hour consumption was 32,640 kw.-hr., the billed demand 161 kw., the actual power factor 36.6 per cent, and total charge, including energy cost, demand charge and coal factor addition, was $1035.67. The billed demand was (161×80) divided by 36.6, or 351.9 kw. and accordingly the demand charge

200 kw. @ $2.00 per kw. per mo.	$400.00
151.9 kw. @ 1.50 per kw. per mo.	227.85
making a total of	$627.85

Had the power factor at that time been up to standard, that is, 80 per cent, the billed demand would have been equal to (161×80) divided by 80, or 161 kw. and the demand charge $322.00. In reality, because of the fact that regardless of demand or energy consumption the first 200 kw. demand must be paid for, the actual demand charge would have been $400.00 instead of $627.85, thereby resulting in a net saving of $227.85 on an otherwise net bill of $1035.67 which, based upon an interest rate of 6 per cent, represents an investment of approximately $45,600.00, or an amount equal to five times that required to remotor the plant and install a corrective device capable of maintaining a power factor of 100 per cent or unity.

According to conditions existing at the time, the unit cost of energy was 3.1 cents, while if the power factor had instead been 80 per cent, this charge would have been 2.1 cents, thus resulting in a saving of about 31 per cent.

Analysis of another and later power bill discloses interesting information showing not only the greater saving effected by a still higher power factor, but also the effect of the demand clause of the company's contract with the utility upon an otherwise reduced power bill.

According to the November, 1922, statement, we have:

Kilowatt hours	23,400	
Reactive kilowatt hours	29,520	
Measured demand	146.8	kw.
Average power factor	56.6	per cent
Standard power factor	80.0	per cent
Demand — $\frac{146.8 \times 80}{56.6}$	207.5	kw.
Billed demand—75 per cent of 396.9 kw. (which is Feb., 1922, demand)	297.5	kw.
Demand charge		
200 kw. @ $2.00 per kw. per mo.	$400.00	
97.7 kw. @ 1.50 per kw. per mo.	146.55	
	$546.55	
Energy charge		
23,400 kw.-hr. @ 1.25 cents	$292.50	
	$839.05	
Discount	9.39	
	$829.66	
Coal factor addition	39.31	
Net	$868.97	

Assuming that instead of 56.6 per cent, the power factor had been as in August, 1921, 31 per cent, the billed demand would have been (146.8×80) divided by 31, or 378.8 kw., thus resulting in a demand charge of

200 kw. @ $2.00 per kw. per mo.	$400.00
178.8 kw. @ 1.50 per kw. per mo.	268.20
	$668.20

Under these conditions the saving is then equal to the difference between $668.20 and the actual demand charge of $546.55, or $121.65.

It must, however, be remembered that the demand charge actually made was based upon a billed demand, as indicated above, of 297.7 kw. instead of (146.8×80) divided by 56.6, or 207.5 kw., as would have been the case had it not been necessary to meet 75 per cent of the February demand. As a consequence, the billed demand would have been

200 kw. @ $2.00 per kw. per mo.	$400.00
7.5 kw. @ 1.50 per kw. per mo.	11.25
	$411.25

or a net saving of $668.20 less $411.25, or $256.95.

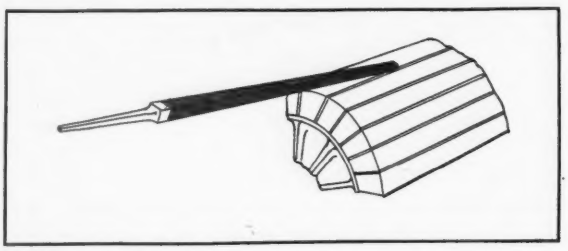
The necessity of reducing the billed demand and, if possible, the elimination of a fluctuating demand is thus obvious. The latter is, of course, difficult of attainment, but the former can be accomplished by maintaining a maximum degree of power factor and when such can be done by a modification of the primary fixed demand charge through a change of contract. It is hoped that when a new contract is made the public utility company will either reduce the fixed demand of 200 kw. or at least the charge for this first 200 kw.

Pointers on the Maintenance of Commutators

PRACTICAL HINTS, WHICH IF FOLLOWED WILL REDUCE REPAIR AND MAINTENANCE COSTS. BY A. R. KNAPP

WHEN WORD COMES to the man in charge of the electrical equipment of a plant that a motor is not acting properly, his thoughts will in all probability immediately concentrate on one particular part of that machine. That part will be the commutator. A large percentage of everyday motor trouble can be traced directly to that important member. It may properly be termed the business end of a motor, and it is good business to see that it get the best possible care and attention.

Most textbooks on electrical machinery usually give a lengthy list of probable causes of motor troubles. It is the purpose of this article to give a few hints on the care of commutators from the viewpoint of the man directly charged with the responsibility of continuous operation. All of the cases and remedies mentioned herein have been encountered in actual practice, and it



SKETCH SHOWING METHOD OF UNDERCUTTING COMMUTATOR WITH FILE

is the earnest hope of the writer that some of them may be of benefit to the craft.

To insure that smooth glossy chocolate color to commutators so much sought by operators, several simple rules must be formulated and strictly adhered to. Rule one is that a regular daily inspection be made of all motors. It is only in this way that one can keep in touch with each motor's peculiarities, and promptly check any irregularity that may arise from day to day.

CLEAN MOTORS WITH COMPRESSED AIR

It should be the rule to clean motors periodically. Compressed air is best for this purpose. The armature core should be thoroughly cleaned. Especially the ventilating air ducts through the core, which are put there for the express purpose of creating a current of air through the core, thereby carrying away much of the heat generated in the armature windings. Yet it is no uncommon occurrence to find these ducts completely clogged with solid matter.

With motors operating in dusty places, particles of dust and the products of manufacture will often collect on the brush holders to a considerable extent. Should there be sparking at brushes, this foreign matter often catches fire. This is not only dangerous to the motor, but the fire hazard to the building is greatly increased, due to flying sparks.

Rule two is that all commutators be wiped clean with a soft cloth each day. This is a very simple operation, but a very important one. Ninety per cent of motor troubles originate from neglected commutators that are allowed to get gummy with oil leakage from the motor bearing. Together with carbon and copper dust it will form a gummy smear on the commutator surface, the brushes become saturated with it, and from that point it is but a short step to costly repairs.

This mixture of carbon and oil is a conductor of electricity, and it will eventually soak into the mica segments, resulting finally in leakage of current from bar to bar. A small arc will be set up between the two bars. This arc may burn itself free, but in the majority of cases it will increase in intensity, resulting in a burnout between bars. Frequently good armature coils are ruined from the fusing together of two bars.

A clean commutator and brushes, kept dry and free from oil, will very rarely shown signs of burning between bars. Eliminate this hazard, and the operator has taken a great step toward reducing to a minimum costly motor shut downs and repairs. Therefore, make it a positive rule to keep every commutator on the job perfectly dry and clean.

Sandpaper has no logical place in the operation of commutators, and the farther away it is kept from the smooth surface of a mass of copper and mica, so much the better for motor harmony. The same may be said of commutator compounds—that black gummy stuff so often passed on to the man in charge as a cure-all for motor troubles. These compounds are largely a lubricant, and if lubrication is needed use paraffin or a small amount of machine oil applied with a cloth.

Frequently troubles arise due to other causes than the one mentioned above. Glowing of the brushes at the surface of the commutator sometimes occurs. This can be due to a number of causes, such as incorrect brush position, brushes not properly spaced, high mica and brushes covering too many bars. One serious case of glowing was corrected by substituting brushes $\frac{1}{8}$ in. narrower than those in use.

Sparking due to burnt spots on the commutator is a very common occurrence. These spots may be equally spaced, or they may occur at irregular intervals. In either case, the cause of this particular trouble is often hard to locate. In one instance, it was found to be caused by a loose coil connection to the commutator. The trouble was further aggravated by the fact that this loose connection was what is known as a "flying open." Violent sparking would occur when the motor was in operation, then suddenly the sparking would cease. This would go on intermittently during the day. The trouble was finally traced to the loose armature lead which would make and break contact, probably due to variations in the motor speed.

EFFECTS OF OPEN CIRCUITS

Effects of a dead open circuit are very apparent. There is a violent snapping bluish-green spark at each brush edge, as the bar of the open-circuited coil passes beneath each set of brushes. The spark seems to reach out from the brush edge, and attempt to follow around the commutator. The bars to which this coil is connected will be found to be burned and pitted, and the mica segments between eaten out to a considerable depth.

In another case, black spots were found to be caused by one set of brush holders moving out of proper alinement. A strip of heavy paper was drawn about the commutator, and the brush positions marked. The paper was then withdrawn, and the marks measured. They were found to be unequally divided. The strip was then marked off in four equal divisions, and the brushes set to these marks.

An overloaded motor will often produce burnt spots on a commutator. A frequent cause of burning is due to high mica. All, or nearly all the bars will be found in this condition, leaving the mica segments protruding slightly above the surface. Continual sparking and heating results, due to poor brush contact. If this condition persists for any length of time, the commutator will become hot enough to melt the solder from the coil connections. A shut-down and a repair is then a certainty.

High Mica

In some cases, high mica is the result of several conditions in no wise chargeable to the operator. One of these is that the mica is of too hard a grade. The copper wears faster than the mica, leaving it protruding above the surface of the copper. Another is that the copper bars are considerably softer than the mica. This condition will produce the same result. Segment copper is usually drop forged or hard drawn. With hard drawn bars, the copper and mica will usually wear evenly. It is with cast copper bars, formerly largely used, that most of the high mica troubles arise.

Most of the men in charge of electrical equipment will work out their own salvation when troubles arise. Some will be content to turn down the commutator occasionally, and let it go at that. This method will give a smooth running machine for a short time, but it is costly in labor and greatly shortens the life of the commutator. Some of them will be guided by the opinion of a smooth-tongued brush salesman. This pest will usually advise their "special abrasive brush made expressly for such conditions." He goes on to state that the abrasive qualities of the brush will grind down the high mica, and your troubles will end as if by magic. He fails to state, however, that the brush will also grind down the copper and very decidedly shorten the life of the commutator.

A man that is really onto his job will seek until he has located the cause of the trouble. Once the cause is found and remedied, sparking will cease permanently.

If it is certain that high mica results from the natural condition of the copper or mica, then undercutting is about the only permanent relief. To do a good job of undercutting requires some knowledge of operating conditions, and quite a bit of patience. Many different tools and methods have been devised to do the work. The average manufacturing plant would not be justified in purchasing one of the elaborate electrically driven saw outfits made for this purpose. Occasionally the work can be done in a milling machine. All things considered, the hand method, while taking longer to accomplish the same result, nevertheless often produces a more satisfactory job. A 6-in. slim taper file is the only tool necessary. Break off about ½ in. of the file tip, and grind square. Hold the file at a slight angle to the commutator surface, about as shown in the accompanying sketch.

Start the file at the end of the mica segment and work back toward the armature core, keeping in the center of the mica. The square end of the file will act as a sort of plow. Remove the mica to a depth of about 1/32 in. The shallower the slots, the easier to keep clean and free from accumulations of carbon dust and grit. Be sure that the entire width of the mica is removed. A thin sliver left flush with the surface of the copper will often defeat the purpose of undercutting. All sharp edges should be removed from the copper bars.

One precaution to be taken with undercut commutators is that oil should not be permitted to work onto the commutator. Oil and carbon dust will clog the slots and cause endless trouble.

A hard non-abrasive carbon brush will give good results when undercutting is resorted to. Brush tension can also be reduced considerably. Usually 1½ lb. per sq. in. of cross section is sufficient.

Practically all repairs to commutators can be made in the average industrial plant repair shop.

Teaches Wife First Aid; She Saves His Life

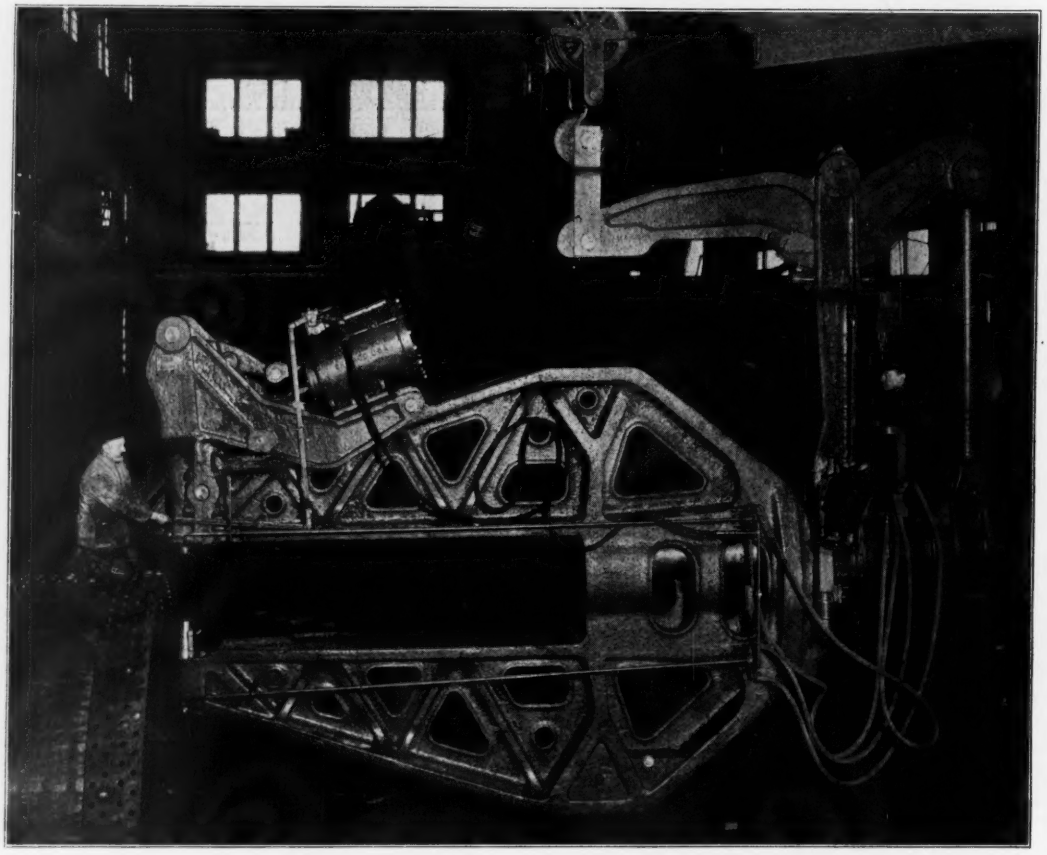
Presence of mind and a knowledge of first aid methods enabled Mrs. F. L. Stanley to save the life of her husband, a patrolman for the Texas Power and Light Co., after he had been rendered unconscious by a discharge of 60,000 v. The accident occurred on a lonely road 15 mi. from Dallas. Patrolman Stanley, wishing to measure the distance of a high voltage conductor from the highway, threw a linen tape, which he supposed was a non-conductor, over the wires. The tape, however, had very fine metal threads woven into it and Mr. Stanley was stunned by the discharge, his left arm and both feet being badly burned. Mrs. Stanley, who accompanied her husband on the inspection trip, turned him over on his stomach and applied the prone pressure method of resuscitation. As soon as he had recovered consciousness, she helped him to the automobile. Mrs. Stanley had never driven a car before, but with her husband's direction she managed to drive 2 mi. to the nearest town where she phoned for an ambulance. She then collapsed from the terrible strain.

Mr. Stanley owes his life to the courage and presence of mind of his wife and to the methods of first aid taught at the safety meetings of the Texas Power and Light Co. Every patrolman was examined as to his knowledge of the prone pressure method and all were recommended to teach the method to their wives, who frequently accompany them on their trips. Patrolman Stanley did so, little realizing that his life would soon depend upon it.

Mrs. Stanley has been recommended for the Samuel Insull Medal, which is awarded for life saving by resuscitation.—National Safety News.

Daily production of electricity by public-utility power plants for the month of June was 151,000,000 kw.-hr.—practically the same as the revised figure for the May rate. This indicates that the increase in the demand for electricity at the present time is counterbalancing the usual seasonal decline in demand, which usually occurs during the summer months. The average production of electricity during April, May and June apparently indicates that there will not be much change in the demand during the balance of the summer; the tendency is towards an increase rather than a decline.

Huge Portable Riveter Used by Niagara Falls Power Company



FOR the erection of the casings and penstocks for the 70,000-hp. hydraulic turbines being installed by the Niagara Falls Power Co., many unusual problems have had to be solved due to the size of the parts.

For instance, in order to handle the riveting of these parts it was necessary to construct the largest portable riveter which has ever been built. This machine, which is shown in the accompanying illustration, weighs 57,750 lb., has a 118-in. reach and will exert a pressure of 150 T. in setting a rivet.

It is evident that mechanical power is necessary to move this riveter into the operating positions. The supporting mechanism is designed so that when revolving the frame on the spindle or tilting it upward or downward from the horizontal, the center of gravity of all the parts hanging upon the crane hook is neither raised nor lowered. With this arrangement, the friction of the bearings and the inertia effects are the only forces to be overcome. These two motions, revolving and tilting, are performed by two reversible, close quarters, air drill motors of approximately 2 hp. each, manufactured by the Chicago Pneumatic Tool Co. The motor for tilting is mounted upon the inner vertical (tension) link and drives a worm gear which engages with a worm gear sector upon the suspension beam. The tilting motion is limited to 30 deg. below the horizontal. The motor for rotating is mounted upon the spindle housing and drives a full worm gear upon the frame. The machine was built by the Hanna Engineering Works of Chicago.

Three of these 70,000-hp. turbines are being installed in the new plant, two of which were built by the Wm. Camp and Sons Ship & Engine Building Co. and the third by the Allis-Chalmers Mfg. Co. The greatest distance across the volute of the casing of each unit is approximately 48 ft., and the diameter of the entrance for water into the volute is 15 ft. The plates vary in thickness from 1¼ in. at the entrance to ⅞ in. at the end of the volute, with rivets of diameter from 1½ in. to 1 in. and 7800 in number. The longest grip for the 1½-in. rivets is approximately 5 in. The penstock is approximately 18 ft. in diameter, with plates 1½-in. thickness, and is about 110 ft. long. Three such penstocks are to be riveted.

Mechanical Balance of Rotating Machinery

By Philip G. Bernholz

MECHANICAL BALANCE is a problem that must be faced by all manufacturers of rotating machinery and is an item that adds considerably to the cost of manufacturing as well as an item that has much to do with the satisfactory operation of a machine and of making a satisfied customer. Of the regular commercial machines, very few are balanced as well as they should be, or so as to be practically without vibration when run on a yielding foundation; the general practice being to pass them if they do not vibrate when standing on a solid iron floor.

While I have had experience around all kinds of machines, including large steam turbines, I will devote most of this article to the medium size and medium speed machines as most commonly used in the commercial field of electric drive.

As the induction motor is without a question the most common machine as well as a good example for illustrating what I wish to state, I shall take that machine

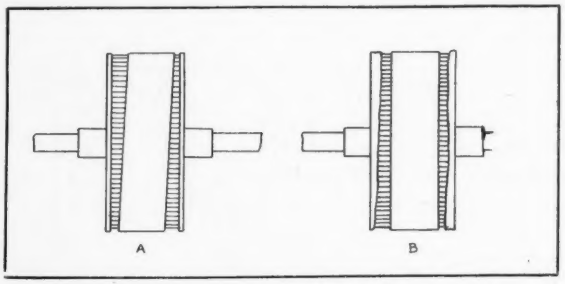


DIAGRAM SHOWING TWO EXAMPLES OF UNBALANCE IN ROTORS

first. At the present time, most induction motor rotors of the medium sizes have the end rings cast on or the entire rotor winding including bars and end rings all one casting. Such cast-on rings are usually rather heavy and often trimmed on only two sides in a lathe which may leave the rings considerably out of balance. Experience has shown that such rotors are much more often out of balance and much harder to balance than the old style rotors where the end rings were made up of copper punchings or strip copper soldered or riveted to the bars. Such rotors are generally first balanced on a pair of horizontal rails, giving them a static balance which is generally good enough to make the machine passable at ordinary speeds; occasionally, however, a rotor so balanced will run very roughly and further balancing on the rails proves little better, if any at all. If such a rotor is to be used for a high-speed machine or on a steel base that has any spring effect, considerable time and expense may be added before any decent balance is obtained if the problem is not taken hold of in the right way.

In the accompanying diagram are two examples of rotors that are likely to give the most trouble. The sketches of course are exaggerated, for in reality it takes but very little unbalance; in fact, so little that it is sometimes hard to detect when the rotor is placed in a lathe and carefully examined. Rotors that can not be balanced well, after one or two good rail balancings, should be placed in a lathe and very carefully checked to see that the core runs true as well as the end rings, flanges, etc. If the core is not true, it should be straightened, if possible. For correct balancing the weights must not only be placed directly opposite the heavy side of the rotor, but also proportionally in the lateral direction; that is, if the heavy part is all in one end ring, the counter weight must all be placed on the same end of the rotor. The rail balance will not determine this or any of the foregoing troubles and, if an inspection does not reveal it, then it must be left for a running balance to determine.

Taking Running Balance

In taking running balance, nearly everybody has his own ideas and ways in going about it; but the one who can combine sound thinking with experience generally gets results where others fail. Nearly all after a while resort to putting whiting on the shaft and making pencil markings while the machine is running to determine the heavy side. While this method is a good one, it is not always easy to tell what the markings mean and in spite of them the weights are placed in the wrong position. In taking such markings, it must always be borne in mind that the markings may mean quite the opposite of what they seem to indicate, depending upon the speed of the machine, stiffness of the shaft, position of the bearings, etc. In one particular case where men had worked for some two days to balance a direct current armature of about 50 hp. at 1200 r.p.m. by the preceding method of marking the shaft, it was later discovered that the markings never came on the heavy or on the light side but were displaced about 50 deg. from the heavy side. An expert was called in on the second day and seeing the condition, ordered the armature to be marked first in one direction of rotation and then in the opposite direction of rotation. The heavy part of the markings then were about 100 deg. apart. The reason for this trouble was due to the armature being run in two pedestal bearings standing on a shaky wooden floor, driving power being transmitted to it through a small belt running on the commutator. When the armature was up to speed, it had considerable vibration and tended to carry the bearings along with it. The bearings with the pedestals being quite heavy retarded the motion and hence the lag. The armature was then balanced good enough to pass in less than half an hour. On low and medium speeds, the markings generally appear on the heavy side while on high speeds they often appear on the light side due to the armature finding its own center of gravity which takes place in all cases after a certain speed which depends upon the flexibility of the shaft.

Marking is very confusing when striking a condition such as is shown in the accompanying diagram in Fig. (A) or (B). For such a case good judgment must be used, as simply shifting weights around as most balancers would do seldom brings good results, but generally winds up by passing the machine slightly better than at the start.

Any such rotor, if well balanced on the rails, can not be much improved by any single weight, as any such a weight while it might seem to improve one end, must of necessity throw the rotor as a whole or static balance out. The only thing that can really cure such a condition is to place two rather heavy but uniform weights on oppo-

site ends of the rotor and 180 deg. apart. This will not disturb the static balance as the one weight will counterbalance the other. Shifting these two weights around, always keeping them 180 deg. apart, will soon show a definite change for better or for worse. After the best location is found it may be necessary to change the amount of weights. I mentioned uniform weights but this might be changed to uniform effective weights as in some armatures it might not be convenient to place the weights the same distance from the center of the shaft on the two ends. In such a case the weights should be inversely proportional to their distance from the center of the shaft; that is, if one weight were placed 6 in. from the center of the shaft on one end and another weight were placed 12 in. away from the center of the shaft on the other end, the latter should be only half as heavy as the former.

Balancing High Speed Machines

On very high speed machines, like steam turbines, all the disks are separately balanced before assembling and the final running balance is carried on in a systematic way with a record sheet showing at a glance every weight, how much, where placed and the results of balance of every important part of the machine given in percentage of balance. The front and rear part of the rotating element are provided for fastening weights and numbered, which numbers are recorded on the test sheet. Each weight is carefully weighed or measured and recorded under the number where it is placed. At each run, the vibration is felt usually by placing the forehead against the handle of a screwdriver which is rested against the bearings, frame, etc., of the machine and the balance of each part of the machine recorded in per cent of balance.

Low Water? Examine Feed Line Valves

By H. A. Jahnke

SWING check valves of the renewable disc type in the feed line from our tilting traps to the boiler make considerable noise when the trap is discharging due to the disc moving up and down. A valve salesman who had considerable experience with the different makes of traps and their connections, recommended a check valve of the type as shown in Fig. 1. The disc of this type of valve is equipped with a piston which moves in a chamber as illustrated. The salesman claimed that this valve would operate almost noiselessly on this kind of service.

I installed one of the valves and found that it made less noise than the swing check. But after a few weeks' use, it was noticed one day that the water was going lower in the boiler. This was puzzling, as I knew that the trap was in operation with the water supply valve wide open. I noticed, however, that the trap tank remained in the dumping position for a longer time than usual, showing that the trap could not discharge the water freely, due to some obstruction in the feed line. I was sure that the feed pipe was clear, hence I was also sure that the new check valve was causing the trouble, as after striking the valve a few light blows with a hammer the trap discharged the water at once, showing that the disc had been held to its seat either by a piece of scale located above the disc piston or the piston was binding in the chamber.

This trouble afterward occurred a number of times, more so when the pressure was high, and we had to strike the valve a few light blows each time to loosen the disc, at times it seemed as if it was necessary to shut down the boiler due to the sticking of the disc and not being able to release this for a time.

At the first opportunity it was possible to leave the pressure out of the boiler, the new check valve was removed and replaced by the swing check again. Since trying the new check valve and having the trouble with it, I am under the impression that a check valve of the swing type is about the best to use where traps are in use for boiler feeding regardless of the noise they make at times. I prefer those of the removable disc type, as I find that they are easier to keep tight; all

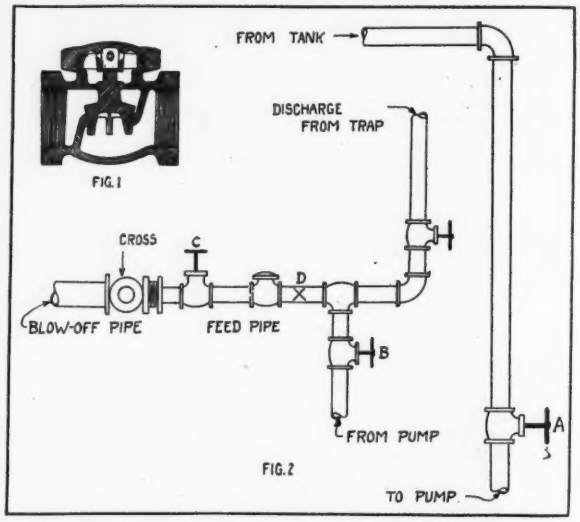


FIG. 1. CHECK VALVE OPERATED NOISELESSLY BUT STICK AT TIMES

FIG. 2. SERIES OF LEAKY VALVES INTERFERES WITH PROPER WATER SUPPLY

that is necessary is to renew the disc in time so as not to cut the seat.

Stop and Check Valves Leaking in Feed Line

In a steam plant where they are using tilting traps for boiler feeding it was noticed for a number of days that the boiler needed much more water than usually with the same load and the trap was just about able to furnish sufficient water with the water supply valve wide open. It also was noticed mornings that there was something wrong in the feed line, or some other place, as in this plant the steam is left on the heating system in the factory at night, with a heavy banked fire under the boiler, while the returns from the heating system are delivered to the boiler feed trap by a tilting lifting trap, then back into the boiler. It had been noticed for some time that the water in the boiler was much lower the next morning than usual, with both traps in good working condition and no condensation going to waste during the night due to leakage in the piping system, etc. But one thing was sure—there was water

leakage somewhere in the feed line which could not be located easily and was discovered only accidentally.

It will be noticed in Fig. 2, which shows the discharge pipe from the boiler feed trap, also the discharge pipe from the feed pump which is used only in case there is trouble with the trap, the trap as well as the pump received water from a hot water tank located in a room above the boiler room. One day it was noticed that the supply pipe from the tank to the pump was very hot while the pump was not in operation and valve A in supply pipe and valve B in discharge pipe of pump were closed. This led to the discovery of where all the feed water was going. The check valve in the feed line near the boiler leaked badly, also both valves A and B had started to leak, which in turn allowed the water to back up into the feed line from the boiler then through valve B also through pump and valve A into the tank, night and day, and when the trap was discharging part of the water would go into the boiler and some back into the tank.

It was decided to repair or renew the leaky check valve in the feed line, so at night after the steam pressure was lowered in the boiler and the stop valve C, which is of the gate type, closed, and both discharge pipes from trap and pump disconnected up to the check valve, it was found that both stop and check valves leaked so badly that they could not hold the pressure of the water in the boiler. Hence the check could not be removed from the line to replace it or repair, for there was no telling how badly the stop valve leaked. As there was not the time to cool off the boiler and drain the water out of it entirely, it was decided to get out of the difficulty by placing another check valve next to the old check as shown at D. Valve B being of the gate type could not be repaired quickly so was disconnected and replaced with a globe valve that was on hand. The leaky check valve was removed and a new stop valve placed at the next boiler cleaning when the weather was more favorable and no danger of the plant and factory freezing while repairs were being made. The valves in the pump were repaired also as they leaked badly.

A Roadman's Experience with a Revolution

ENGINEERING SKILL OF SAM TURNS TIDE OF BATTLE AT PSYCHOLOGICAL MOMENT. BY GEO. H. WALLACE

'WAS I EVER in Mexico? Well I'll say I was, and I came so darned near to staying there that there was no fun in it," was the Roadman's reply to the question, as he sat with the gang one noon on the box in which the governor was packed, while we smoked and told stories of engineering experiences here and there.

"Did they try to run you fer president, or something?" chipped in Hank Norman, the extra man, as he refilled his Missouri briar out of the Roadman's can, handed it back and borrowed a match.

"No, they didn't. But it is hard to say what office they would have handed me if they had caught me—possibly chief interest in a first class funeral. As it was, I didn't have much chance to run for anything for awhile; they brought the excitement right to me and dared me to accept the office, whatever it was.

"And if it had not been for a little Mexican pump man who was on the job and put in about half his time up in the main plant watching me set up the engine, those revolucionarios would have had me so punched full of holes that you could have used me for a coal screen."

Jim Beezley waited about as long as he could stand it, and then said, "Well, what did they do to you?"

"What did they do? Listen," and the Roadman consulted his watch, "there's time enough, I guess, before the whistle blows, if none of you Hombres don't butt in too much. Emmons, you sit down and shut up. Time is limited. Here you, Wheatley, get your feet off that metallic packing, what do you think it is—a cuspidor, or something? Now all hands, shut up.

"It was a number of years ago, when revolutions were apt to crop out down below the Rio Grande just about every so often, or oftener, like fire sales in the lower east side in New York. About the time that one general got the country hog tied so that he thought it safe to sleep all night without getting up two or three times to look in the clothes closet for a *Cabecilla*, some other buck private would promote himself to the rank of general (there's nothing between a private and a general down there) and start another revolution. And if he was so fortunate as to catch his immediate predecessor on the job, he would stand him up against a wall and punch him full of little holes and take over the throne. When one of these ambitious *Hombre de bien* did get a cinch on the presidency, he at once took the census of all that were suspected to be on the opposite side of the argument, and picked out a few for sacrifice just for moral effect.

"I had been sent down there to erect a new engine at a stamp mill located at a mine so far away from civilization, that we got our mail once in two weeks, if some bandit gang didn't get it sooner. The location was ideal for everything not pertaining to work. The only word you could hear was 'manyana,' and if you wanted some of those greasers to do anything, it was always 'manyana.' They were the originators of the expression, 'Never do today what you can put off till tomorrow.'

"It was quite a job, and we were getting along pretty well when you consider that all the Spanish I knew had been picked up on the job, and all the English that the gang ever knew had been forgotten the day before.

"This plant was not a new one. The old engines and boilers were still there, but pretty well worn out. The power house had been added onto for housing the new equipment. Oil was the fuel and it was pumped to the plant from Loredillo, a number of miles away, and stored in a couple of tanks on a hill some little distance from the mines proper, a distance that we later found was just out of rifle range as far as the revolutionists were concerned.

"Some distance from the power house was a river where was located a pump house in which was on old fire-box boiler and a pump which supplied the water for

the plant and the mine. There was a tunnel leading to the pump house in which the pipes were laid and which played a leading part in the war later on.

"This pump was the pet of Sam Rodriguez, the little Mexican I spoke of; but as I said, he put in most of his time up with me, as the oil fuel needed little attention. We had a motor driven pump at the job to replace this outfit and we were in hopes that the old boiler there would stay in one piece till we got the motor installed.

"On each side of the pump house were low open sheds covered on the rear with corrugated iron partly rusted through in places, in which were kept a few tools such as shovels, wheelbarrows, some cement and other junk, including a box of dynamite.

"Owing to the prevalence of bandits in the country, the mining company had supplied us with a number of Krag rifles and plenty of ammunition. They are good guns and proved themselves so that day. The mine had

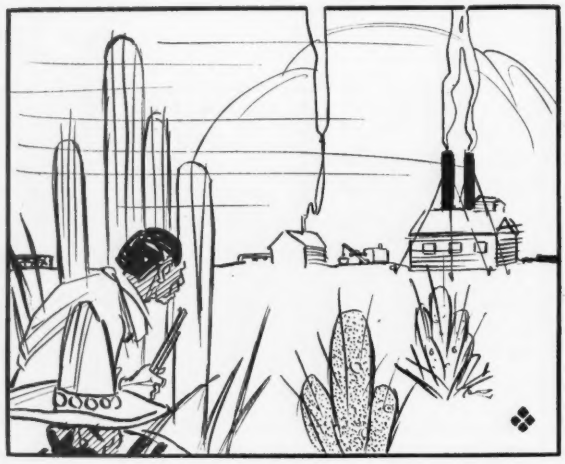


FIG. 1. THEIR FIRE HAD SLACKENED SOMEWHAT AND IT SEEMED THEY WERE READY TO CALL THE WAR OFF

been shut down for some time owing to the general run down condition of the old plant, and there was but a small force of clerks in the office near the plant, young bloods, soldiers of fortune and boomers looking for excitement. They got it.

The Revolutionists Approach

"We were just about ready to turn steam on the engine and try her out. The boiler erector was on the job, old Sandy McGavin, a veteran of the South African war who was always saying that the place was dead and there was no excitement going on. He changed his mind also. We were finishing up the little odd jobs one day and expected to get going on the next, when little Sam came in from the pump house all out of breath and yelled,

" 'Capitan, mucho solado, alma de Dios, Sacramento sapolio.'

"I ran to the window and looked out in the direction where he was pointing with his sombrero and sure enough, there was quite a formidable little army just beyond the tanks on the hill taking stock of us. They fired a few shots that fell short, all but one. It must have struck a stone and glanced, as it came through the sheet iron sides of the engine room and struck the brick wall and fell to the floor. It was one of those old 0.45 Springfields, which I suspected when I saw the black powder smoke.

"One of the boys ran to the door and gave the alarm, and the clerks came out of the little wooden office and reinforced us. Fortunately the power house hid their movement from the army on the hill on the opposite side of the plant.

"Several of these clerks were old ex-regs of Uncle Sam, and knew how to handle a rifle. Bill Shiflet, I remember, had been in the service 12 yr. Besides, there were some others who preferred to say little about their past life.

"I issued rifles and ammunition all around and posted the men to the best advantage, feeling that if we were to have a war, we might as well have a good one. We fired a few shots—very few—to create the impression that we were few in number, and scattered them a lot, though we did bring down a couple of horses, and a man or so.

"This went on for an hour or so, when I could see the enemy through my field glass pointing to Sam's pump house. They dismounted and leaving a few men with the horses, advanced in open order, slowly making their way down the hill to where they would have the pump house and sheds between them and us, where they hoped to pick us off at their leisure.

"In fact, that is what I hoped they would do. I got some of the clerks to bring out the steel plates that covered the pipe tunnels, the other floor plates in the power house and stood them up against the wall filling in between them with bags of cement that we had in the boiler room. While these plates were but ¼ in. thick, they stopped a number of bullets that were low enough to do damage; most of them, however, went high.

"As the enemy advanced, the affair assumed the proportions of a *Batella Campel* and we were able to pick off more of them, but we were holding our fire till they got closer so as to make a good job of it and have it over with, once and for all time.

"Then we missed Sam. He was nowhere to be seen. Thinking that he had cold feet and had beat it for the basement while the going was good, I dismissed the matter and went on with the merry little war.

"Soon some of the Greasers got the shed between them and us, ran to cover and began shooting at us through the holes that nature had put in the sheet iron. This was pudding for the boys inside and we shot at every puff of black smoke or just below it. We opened up now with all of our reserves and the enemy soon found that they were in a box that they could not get out of.

"Evidently they made signals of distress to the bunch on the hill, for they mounted and leading the extra horses, made a spectacular charge down the hill to the rescue of their comrades. I let them come in a straight line to about 300 yd. and then gave the order to transfer the fire to the cavalry and get a man at every shot.

"We let fly with those Krags in one staggering volley. It was fine shooting and the riders were about annihilated. 'Now for the horses, boys,' I yelled, and we cleaned them out in a few volleys. This little side war sort of put a crimp in the reserves of the enemy and left them no way to retreat.

"Then General Oleomargerino came out from behind the shed waving his hat and lustily yelling 'Amigo,' but Bill Shiflet plugged him for a row of apple trees. This put the rest of the bunch in a pretty tight place. Some of them attempted to run away; but as soon as they appeared above the roof of the shed, they were reminded of the fact that it was more dangerous to run away than to stay and fight.

"Just then Sam came running up and tugging at my sleeve said, 'I fix, Senor; soon no more fight, Sacre Diablo,' and he rattled off a lot of Spanish while his eyes shone with excitement. 'How you fix, Sam?' I asked. 'You see. Pretty soon Militar de Alfenique go way damn pronto,' and he picked up a rifle and began to give it healthy exercise.

"I began to notice that most of the survivors were beginning to gather around the pump house proper where they no doubt thought that the boiler would protect them. This made it equally difficult for us to find them. Their fire had slackened considerably and it seemed that they were about ready to call the war off for the day; but, on the other hand, while we had had two killed and two more slightly wounded, including McGavin, the boys were just getting well interested in the game.

"The sheet iron sides of the power house were pretty well perforated and looked like a nutmeg grater, where those lead slugs came through, but most of them went just high enough to miss us, though all of us had some close calls.

"I had climbed up on the feed water heater to take a peek out of an upper window and see how the war was going on when there was a roar like a siege gun, and we were bombarded with picks, shovels, and wheelbarows, while the shed and pump house disappeared like a busted balloon. That boiler and the dynamite did a fine job of wrecking.

"When the dust cleared away, we saw a few, very few, of that 'noble 600' headed for the hills. Some of them got there and some didn't. Sam came limping up with blood streaming down his face and holding his leg and said: 'I fix, Senor; I tell you I fix.'

" 'Yes, I guess you did, Sam; but what did you fix—blow up the magazine?'

" 'No, I go down pump house in tunnel. Stop pump, turn on more oil and plug safety valve. Boiler he go bang.' "

Just then the whistle blew.

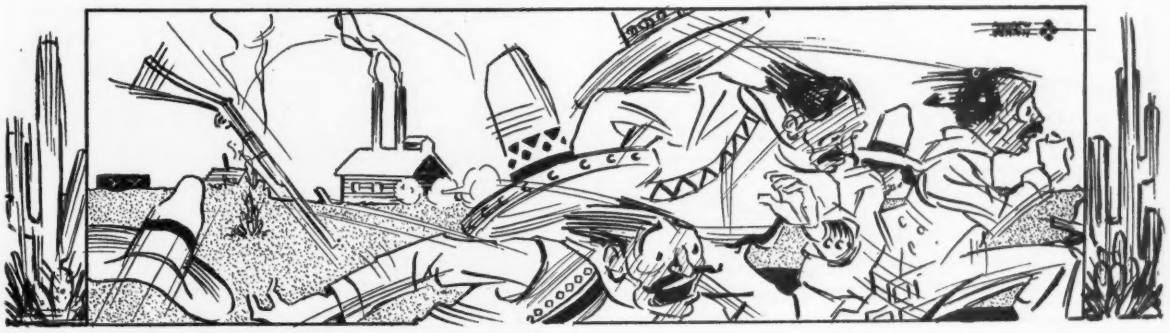


FIG. 2. WHEN THE DUST CLEARED AWAY, VERY FEW OF THAT "NOBLE 600" WERE HEADED FOR THE HILLS

Classification of Fuel Oils

AS ordinarily understood, fuel oil is crude petroleum from which the lighter products such as gasoline, naptha and kerosene have been extracted. Crude petroleum lends itself to two broad general classifications, the oils with a paraffine base and those with an asphaltum base. After the refining process a paraffine base oil yields paraffine as the final product and similarly asphaltum is the final product from the oils of the second classification.

Paraffine base oils are not generally used for fuel for the reason they yield lubricating oils of excellent quality. The heavier asphalt base oils do not yield so well in lubricants and in some cases their yield of naptha and gasoline is so low that the crude oil is used for fuel without being put through the refining process. All mineral or crude oils consist of a mixture of a number of hydrocarbons. These oils all have the same component parts, their differences being due mainly to the varying proportions of their hydrocarbon constituents. Asphaltum base oils are generally heavier gravity oils.

It is customary to sell and describe oil as being of a certain "specific gravity" or of a certain "degree Baumé," the latter has come into common usage for the designation of those oils which are used for fuel. The specific gravity scale is that in which the relation of weights is expressed as the decimal obtained by dividing the weight of a quantity of oil by the weight of the same quantity of water. For the Baumé scale the gravity of water is taken as 10 degrees and the gravity of other liquids are expressed in degrees above and below 10 as the liquid is lighter or heavier than water. Liquids heavier than water are expressed in degrees below 10, and those lighter than water in degrees above 10.

Should it be desired to change degrees Baumé to specific gravity or vice versa, the following formulas will be found applicable for all oils and other liquids which are lighter than water.

$$\text{Specific gravity} = \frac{140}{130 + \text{Baumé reading}}$$

$$\text{Degrees Baumé} = \frac{140}{\text{specific gravity}} - 130$$

Light oils are thin and will flow readily, whereas the heavy oils, those below 20 deg. Baumé, are thick and flow with difficulty.

Constant Excess Pressure Governor

NECESSITY of excess pressure governors in conjunction with boiler feed service is well known to the boiler room operator, but it is not always possible to convince the management of this fact.

Due to trouble experienced with boiler feed control, I made up the apparatus which is shown in Figs. 1 and 2. My trouble was experienced not with the Terry governor or control, but with the constant pressure governors which had been purchased at the same time the turbine driven feed pumps were installed.

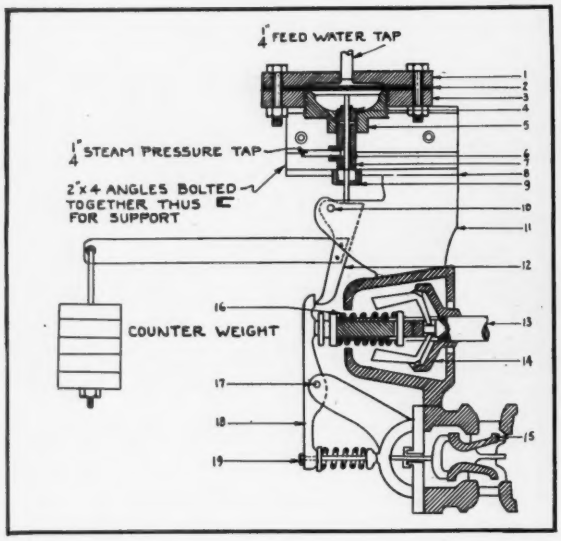


FIG. 1. SKETCH SHOWING SECTION THROUGH TURBINE GOVERNOR, VALVE AND EXCESS PRESSURE GOVERNOR

DESIGNATION OF CENTRAL PARTS

1. Standard 4-in. blind flange, tapped for ¼-in. pipe.
2. Rubber diaphragm.
3. Malleable disc supporting diaphragm.
4. ½-in. plug drilled ¼ in., used as a guide.
5. 4-in. pipe plug.
6. ¼-in. brass rod.
7. ⅜-in. pipe size special fitting.
8. Base which supports control mechanism.
9. Packing gland.
10. Pin supporting lever (12).
11. Special fitting which supported old excess pressure governor.
12. Lever which communicates diaphragm movement to turbine control.
13. Turbine shaft.
14. Governor weights.
15. Governor valve.
16. Governor spring.
17. Pin acting as fulcrum for governor arm (18).
18. Governor arm.
19. Governor valve stem.

In Fig. 1 is shown the governing arrangement, together with its connections to the Terry turbine control. All numbers from 1 to 7 inclusive designate parts of the homemade governor and numbers from 8 to 19 inclusive designate Terry turbine control parts.

It can readily be seen that this governor is made up of equipment which is found lying around every boiler plant. An ordinary angle valve can be used instead of the special brass fitting (7). The fitting shown is an old limit screw from which the stem was removed to permit

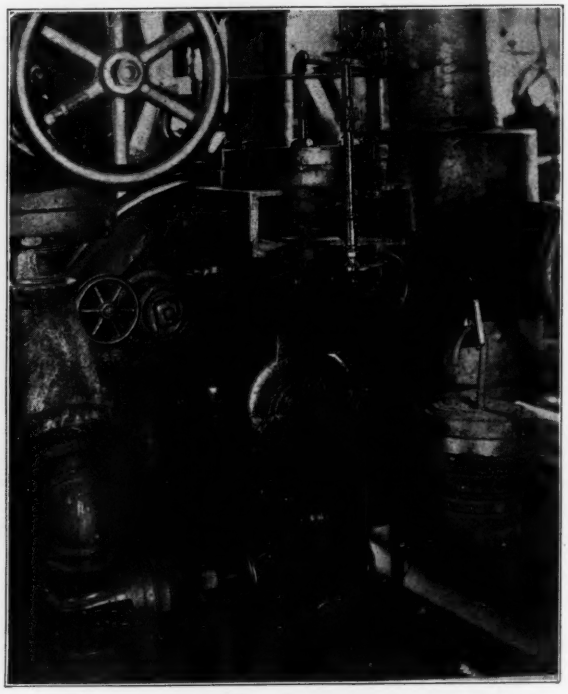


FIG. 2. EXCESS PRESSURE GOVERNOR INSTALLED AND IN OPERATION

the insertion of the ¼-in. brass rod (6). The malleable disk (3) was made by sawing off the end of a 3-in. malleable pipe cap. This was the correct size without any work other than rounding up the edges with a file. The lever supporting the counter-weight was bolted to the lever (12), which was a part of the old constant pressure equipment.

In operation, the rise and fall of steam pressure accelerates or retards the speed of the turbine through the operation of the governor, thus raising or lowering the water pressure and maintaining a constant head above steam pressure. The mechanical action is as follows: A rise in steam pressure raises the diaphragm (2) permitting the upward movement of the disc (3), brass rod (6) and the lever (12) because of the pressure exerted by the counter-weight. The governor arm (18) follows the lever (12) because of the spring on valve stem (19). This opens the governor valve (15) admitting more steam to the turbine which increases its speed and consequently the pressure on the feed water.

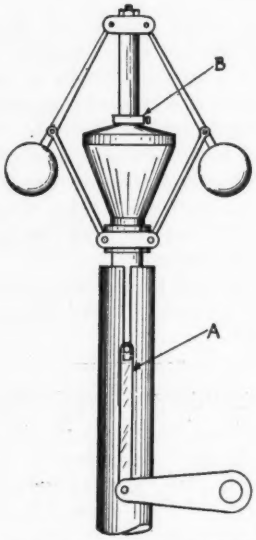
If the pump suction is lost and the turbine starts to over-speed, the counter-weight pushes the lever (12) back, permitting the governor spring (16) and governor weights (14) to function in shutting off the steam supply to the turbine in regular manner.

This governor has been in operation for about 6 mo. and is giving entire satisfaction. Much time is saved the operator in the care of his feed water, besides making it possible for him to devote more time to his fires. It has also smoothed out operating conditions not only in the boiler feed control, but has been an indirect help towards higher efficiency. A differential pressure of from 15 to 20 lb. is maintained, which is ample for the satisfactory operation of the "S-C" feed water regulators.

East Chicago, Ind. C. A. UNDERWOOD.

Faulty Governor Setting

RECENTLY there came to the writer's notice the case of an engine which could not be run on open throttle, as the engine would speed up, and tend to run away. Upon investigation, it was found that the drop



FAULTY ARRANGEMENT OF GOVERNOR

rod, A, as shown in the accompanying sketch, connecting the bell crank to the spindle sleeve, had been lengthened so much that when the arm of the bell crank was in a horizontal position the governor weight had risen to its upper limit; consequently, on a light load the cutoff could not be automatically shortened and the throttle valve had to be partly closed. This was all right as long as the load remained constant; but when a heavy load came on, it was a case of run for the throttle valve.

This rod A was shortened until the arm when standing horizontally was in the middle of its movement up and down and the governor weight was midway between the governor column and the collar B. Governor rods were also changed, so that the hooks would trip when the governor lifted off the stop. The safety cams were also adjusted, so that when the governor was at its lowest position, the hooks would not pick up the valves when the wristplate was at its extreme travel.

TOM JONES.

Gasoline Engine Valve Alterations Increases Fuel Economy

ENGINEERS HAVING Witte engines of the older type might be interested in the way I transformed two engines of that make to a simpler valve action and a hit-and-miss governor, resulting in a saving of 65 per cent of the fuel formerly used.

I took the small fuel valve and the fuel valve lever out, turned the governor end for end on the shaft, took the pin out of the exhaust valve lever roller and in place

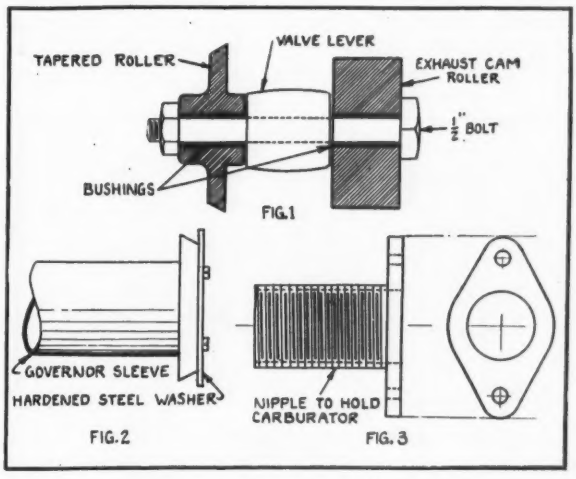


FIG. 1. EXHAUST VALVE LEVER ROLLER
FIG. 2. FUEL VALVE CAM ON GOVERNOR SLEEVE
FIG. 3. NIPPLE FOR ADAPTING FORD CARBURETOR

put in a ½-in. bolt and two bushings (Fig. 1). On one side I have set the tapered cam which was at the end of the fuel valve lever. On the governor sleeve in place of the sharp pointed fuel valve cam, I have put a hardened steel washer (Fig. 2) 3⅛-in. diameter and held in place by the same three screws. When the set speed is exceeded the governor brings the steel washer over the tapered roller now placed on the exhaust valve lever, thus holding the exhaust valve until the governor acts. The bracket collar holding the accelerator fork is now set on the same shaft against the exhaust valve lever and holds it in place.

The fuel feed end I have completely changed by putting on a Ford carburetor which is held in place by a short nipple (Fig. 3), having been shaped with the hacksaw, spread and drilled to suit.

This change was made to a 15 hp. engine. An extra supply of air is taken from the hole which was filled by the small fuel valve and at the nipple connection holding the carburetor. For the 40-hp. engine of the same model I made the same changes, but instead of a carburetor I have put a constant level feed box taken from a 20-hp, Fairbanks-Morse engine. The pump connections were made to fit by slightly altering the pipes; it avoids all the leaks of the old style and gives a better mixture. We use as fuel engine distillate No. 1. The engines are now running much cooler and easier, which also effects a material saving in the amount of lubricating oil used.

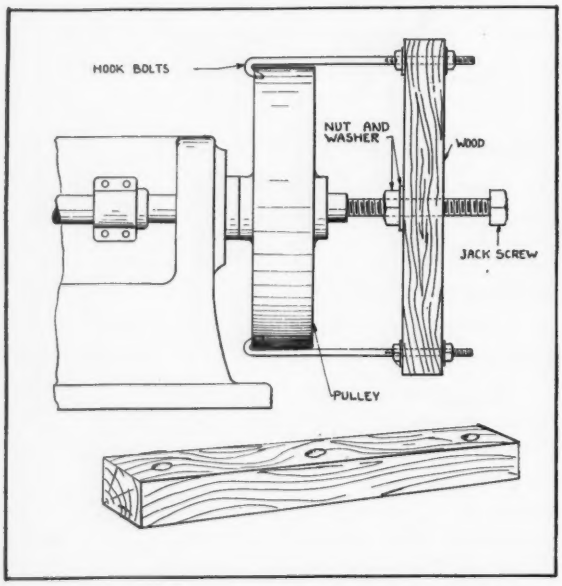
Johnnie, Nevada. CHAS. LABBE.

Wood Serves in Emergency as a Heavy Puller

FOR THE removal of a pulley from a shaft, an emergency puller, the beam of which is made of wood, is illustrated in the accompanying sketch.

In this instance, two long bolts were bent to hook over the flange of the pulley, and secured to the beam by boring holes at the ends and attaching with nuts and washers as indicated.

Through the center of the beam, a bored hole located a third bolt and a nut to thread over the bolt, with a large washer, was the means of obtaining the requisite pressure on the shaft to force the pulley off. This use of a wooden puller can be taken advantage of for the removal of any large force fitted parts. The center screw



ASSEMBLY OF A HEAVY PULLER USED FOR REMOVAL OF A PULLEY FROM ITS SHAFT

with its nut transmits the pressure into the beam as readily as if the beam were of steel and the thread were cut into this. The outside end connections are varied to length of spacing, using bolts or other tie rods, as would be required for any special work. A wooden beam is more frequently available, when a long puller is required, than would be a corresponding part of steel or wrought iron.

Washington, D. C. G. A. LUERS.

Vacuum Breaker on Dashpot Aids in Closing Down

I ONCE had charge of a 20 by 36-in. Corliss engine on which it was impossible to handle both the valve gear and throttle valve at the same time. Since the dashpots were so constructed as to bring the valve gear in central position instantly, losing control of either the throttle or valve gear allowed the engine to stop on center while closing down.

After much inconvenience and loss of time, a ⅜-in. pipe connection was attached to the bottom of the dashpots with valves or stop cocks fitted to the upper ends of the ⅜-in. pipe connections which were made the right height to be convenient to handle. The stops were opened after the engine had half closed down, as the gearing would have to drive down the plungers when the vacuum was broken by the opening of the air connection. With the vacuum removed, the valve gear could be placed in any position and it would remain where set, leaving the operator free to handle the throttle valve. In this way, sufficient time was available to handle either the throttle or gear to keep the engine from centering.

This scheme often came in handy when reversing back and forth to find lost motion and to back up for any other purposes. I have handled several Corliss engines of various sizes, but I have never had dashpots which were so sensitive as on this particular engine. hence this little stunt was a great convenience.

S. P. MORRISON.

Adjusting Cylinder Clearance

IN THE July 1 issue of Power Plant Engineering, M. M. Brown describes a method for adjusting the cylinder clearance of an engine, to which I would like to take a few exceptions.

As illustrated in the article, Mr. Brown shows the liners placed between the wedge and the butt box. This is a wrong place to put the liners on the type of rod shown, because when this type of connecting rod is keyed up, the connecting rod is lengthened. Liners or shims should be placed between the end of the rod and the crown box. When the liners are put in this way, the connecting rod is shortened again, which compensates for the position so that more keying up can be done, without binding the wedges.

If Mr. Brown had placed the liners as described above, he would not have had to screw the piston rod into the crosshead. F. W. R.

Equivalent Pipe Sizes

HUNDREDS OF ideas have been published in the "Letters" and "Problems" departments of Power Plant Engineering which have been of great assistance and profit to engineers. I have often benefited from these suggestions and in return have done my best to help other readers by promptly sending to the editor any idea which in my estimation would be of value. Such an idea came to my notice recently.

It should be of value to all mechanics who work with pipe. Especially is it valuable because it is so readily remembered.

Suppose it is desired to find the size of one pipe which will be equal in area to that of two other smaller ones. As an example, say one is 4 in. and the other 8 in. Draw one line 8 in. long and at right angles to it at one extremity draw another 4 in. long. The distance between the outer extremities of these lines represents the diameter of the equivalent pipe. In this case the distance is slightly over 8 15/16 in., so that a 9-in. pipe would be required. Not more than a quarter of a minute is required to get the answer and a lot of calculation or time thumbing through area tables is saved.

For more than 25 yr. I have been a subscriber to power plant journals and from them I get more real value and pleasure for my money than it is possible to secure for such a small cash outlay in any other way.

Each new issue is likely to contain some particularly interesting and adaptable idea for one who is trying to learn. JAMES E. NOBLE.

Toronto, Ont.

EDITOR'S NOTE: The dimensions referred to in this "kink" should be considered as actual inside diameters and not the nominal diameter of the pipe. There is considerable difference between the two, especially in the smaller sizes.

As it is usually presented, the problem is not primarily one of equivalent areas, but rather is it one of equivalent flow. While in short pipes the flow is approximately proportional to the internal area of the pipe, in long pipes friction plays an increasingly important part and should be taken into consideration. The effect is that two long pipes having a combined net area equal to that of a third will not permit a flow equal to that through the larger pipe owing to the increased surface friction in the two smaller lines.

It may be mentioned here that the method suggested by the author is equally adaptable to problems involving the strength of bolts, stays and tie rods, ropes and cables, solid columns, etc.

Emergency Gaskets

IN THE July 1 issue of Power Plant Engineering, there is an article on the use of packing in place of a gasket, by R. C. Hendee, which brings to mind several instances where substitutes were used as gaskets and in every case served as well as material especially designed for the service.

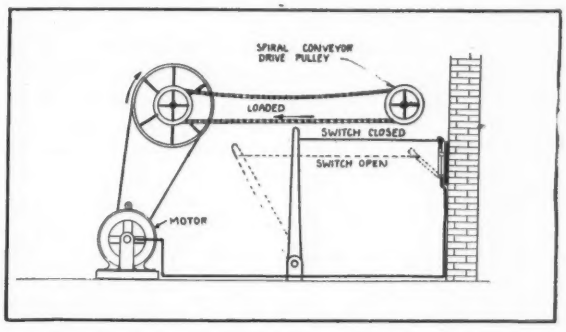
In one case a large steam trap was to be overhauled, new float put in, etc., but when the large cover was to be put on after all repairs had been made, it was found that there was no sheet packing on the job large enough out of which to cut a gasket and as the trap was an important part of a heating system it was necessary that it be put back in service without delay. A gasket was cut from a piece of two-ply tar roofing paper and without the application of "dope" the joint was made up and after years of service, though under only 10 lb. pressure, is still tight and has never been followed up.

At another time, a flange on a main steam line began to leak badly and it was at once evident that the gasket must be renewed before a delayed supply of sheet packing reached the plant, therefore the flange was opened, cleaned off with the hope that the packing would be at hand in time to effect repairs, but this did not prove the case. A section of No. 14 rubber covered copper wire was brought up, the insulation removed from one end, and this bare end was passed through one of the bolt holes, then with all the bolts in place on one side of the flange a flat coil of wire was wound carefully inside the bolt circle, the flange drawn together, and steam turned on while the joint was being followed up. The joint remained tight for many months after following up at intervals. Another case is recalled where two turns of 5/8-in. hemp rope were driven into the packing grooves of a deep well pump plunger and served as well as the regular combination packing.

These instances are, of course, merely substitutes and show, as does Mr. Hendee's application, the resourcefulness necessary where lack of proper material would otherwise cripple a plant and in some cases cause serious and costly shutdowns. SUBSTITUTE.

Stopping Coal Conveyor Automatically

IN OUR plant the coal is supplied to the boiler room by a screw conveyor, operated by an electric motor. The man who looked after the coal supply would trim down a quantity of coal in the bunker to where it would run into the conveyor, and then go to other parts of the plant to attend to other duties. In the meantime the coal would frequently stop running and the conveyor, running light, would vibrate and make much noise. The operator being of an ingenious turn of mind, contrived a device for controlling the conveyor automatically whenever the coal stopped running. Referring to the sketch, the sprocket chain belt was in line with the knife-switch controlling the motor and ran in the



SLACK CHAIN STRIKES LEVER AND SWITCH OPENS

direction indicated by the arrow. When the conveyor was loaded the under side of the chain was taut and just cleared the end of the wooden lever hinged to the floor. When the coal stopped the belt would run slack and strike the tend of the lever and push it over to the left which, being attached to the switch by a cord, would open the switch and stop the motor. Upon starting, after the machine was loaded, the lever was adjusted to function whenever the load would go off.

Camden, Me. M. M. BROWN.

Burning Tires

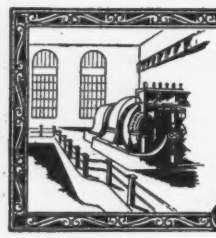
DURING THE greater part of last winter, and especially when coal was scarce and expensive, I got good results in our boiler room by using automobile tires as fuel along with coal.

They are very high in volatile, and are inclined to cause smoke; but if burned with wood or coal, give splendid and intense heat.

One ordinary tire, properly burned with enough extra air to prevent smoke, will equal its weight in ordinary coal. This is a tremendous saving when you can have a ton of tires hauled from a nearby garage or the city dumping grounds for a couple of dollars or less. They are especially good for quick raising of steam, and are a great financial investment.

Winnipeg, Man. C. RYE.

WHEN A boiler is down for repairs, the baffling should always be inspected for small cracks. These should be cemented tightly. Concentration of flow gradually forces a larger opening until failure of the baffle results.



PROBLEMS IN OPERATING POWER EQUIPMENT

Will Two Feed Water Systems Be Necessary?

On page 796 of your Aug. 1 issue, A. C. asks regarding the feeding of boilers operating on different steam pressures with one feed pump supplying water to both high and low pressure units.

For the conditions described, it will be entirely unnecessary to have separate feed pumps. The feed water can be accurately controlled by equipping all of the boilers with regulators of the continuous flow type. Should regulators of the intermittent type be installed, the results would not be satisfactory as the valves controlling the flow would be either wide open or tight shut, consequently the 150-lb. pressure unit would be adversely affected at certain periods of operation. Furthermore, it is absolutely necessary that the control valves of the regulators have areas correctly proportioned for the evaporation and water pressure drop requirements of this individual plant.

The amount of water which may be passed by a valve is determined not only by the area but also by the pressure, therefore, it would be entirely possible for A. C. to maintain 175 lb. water pressure in the main feed header, and by installing regulators on all of the boilers with valves of the proper area, the 150-lb. pressure boiler could be fed on a differential water pressure of 25 lb., and the 120-lb. boilers on a water pressure differential of 55 lb.

To regulate the feed flow successfully and maintain a uniform excess of water pressure over the steam pressures at all times, it is imperative that the boiler feed pump be controlled by a constant excess pressure type pump governor. Thus a uniform water pressure differential is assured, regardless of variations in steam pressures.

A. C.'s problem is similar to that of other plants where new boilers of higher pressures are installed in connection with older boilers. The feed water in these instances has been successfully regulated in the manner described and equally satisfactory results could be secured by A. C. through the installation of the proper type of continuous flow feed water regulators.

Erie, Pa. C. E. Wolff, Jr.

In the Aug. 1 issue of Power Plant Engineering, A. C. wishes to know whether it is necessary to have two separate water feed systems to take care of boilers operating at 120 and 150-lb. pressure.

There must be some reason for working this boiler at 150 lb. with the others at 120 lb., but A. C. states that he intends to place a reducing valve on the main steam header reducing the 150 to 120 to equalize. If there is no process that will require 150 lb., it would be better to operate the new boiler at the same pressure as the others, which would do away with reducing valves.

There will be trouble feeding water to boilers carrying two different pressures as water will flow to the place of least resistance. In fact, when one or more boilers operating in a battery become lower in water than the others they will take the water first and I have seen many instances where a battery of five or six boilers would all have low water at once on this account, although they were equipped with feed water regulators.

Where all the boilers are working about even, that is, when each is evaporating about the same amount of water, it would be possible to feed this boiler with the one system when the pressure is carried high enough on the feed water pump to feed the highest pressure boiler.

North Tonawanda, N. Y. Chas. J. Miller.

Scale in Feed Water Line

G. T.'s problem in the Aug. 1 issue concerning the formation of scale in the feed water line was of particular interest to me because I encountered an almost similar condition of affairs some years ago in a medium sized plant where we were using a fairly bad water, had a closed feed water system, and had a long feed line which continually clogged up with scale and sediment.

We first tried feeding soda ash in the water before it was pumped through the closed heater; but instead of improving things, this made it worse because the deposit in the feed lines was increased, doubtless due to the additional scale forming impurities thrown out of the water by the soda ash. Our next expedient was to have made up a duplicate section of feed line, with flanged connections so that we could keep the line in service while the scaled section was being cleaned out.

Our problem was finally solved by an open feed water heater salesman, who advanced the suggestion that the deposit was largely due to the fact that the impurities were released from the water upon heating, but did not have occasion to settle out before going into the line. We finally made the installation of an open heater, and it immediately relieved us of the pipe line deposit. The incoming water upon being heated in the open heater gave up its carbonic acid gas (which holds some of the scale forming matter in solution), and upon the release of this gas, a noticeable precipitate was formed. The heater settling section was made extra large so that the water would settle there for about 45 min., and twice a day we blew out a large quantity of deposit and mud from the blowoff valves in the bottom of the heater. That the CO_2 gas was being driven off was very readily proven, because if we closed the gate valve on the vent line from the heater it took only a short time for the heater to fill with air and gases, which, upon re-opening the valve, could be heard rushing out. The deposit obtained in the heater in this way not only relieved the scale formation in the pipe line, but also noticeably cut down the mud and sludge in the boiler.

It is my recommendation, therefore, that G. T. give consideration to overcoming his trouble by means of this type of heater installation.

The filter which he contemplates using would be of no service, especially if used before the heater, because at that point the impurities are all in solution and naturally would not be stopped in the filter. If the filter was put between the closed heater and the boiler, it might catch some deposit; but, being subjected to boiler pressure, it would probably be too small to give any really good service.

The fact that the water as it comes from the pond is clear does not, of course, indicate that it is soft; and in view of the scale which is being formed, an analysis of it would probably show considerable hardness.

Philadelphia, Pa. M. A. SALLER.

Use Exhaust Steam

AT A PLANT in which I am interested there is a 60 hp. boiler operating at 80 lb. pressure, and a steam engine, exhausting to atmosphere which uses 40 hp. of the boiler capacity. The plant has a dry room and heating system which requires the remaining 20 boiler hp. In severe cold weather more heating capacity than this is needed.

How much back pressure could be put on the engine so that it would consume the total output of the boiler? Would it be economical to operate the engine on back pressure when only about ⅓ or ½ of the exhaust could be utilized for heating? S. P. E.

A. To secure the most economical operation of a combination power and heating plant, the exhaust from the prime movers should be utilized to the fullest possible extent. There is certainly nothing gained in providing boiler capacity for a heating load and at the same time allowing the engine exhaust to go to waste. By increasing the back pressure on the engine to perhaps 5 lb. gage, adequate exhaust for heating would be available and the engine would be able to deliver the necessary amount of power with but very little increase in steam consumption. It is to be suggested that a back pressure valve, set at about 5 lb. gage, be installed on the exhaust line and a connection made from a point under this valve to the heating system.

If after the heating demands have been satisfied, there is still exhaust steam available it would be well to use this for heating feed water.

If an injector is used for feeding the boiler it will probably have to be replaced with a boiler feed pump for handling the heated feed water.

Maintenance Schedules for Oil Engine Plants

IN THE July 1 issue, R. H. requested information regarding the laying out of maintenance schedules for oil engine plants. Comparatively little information appears to be available on this subject, although it is receiving more attention each year.

Experience has shown that smooth running, maximum service, and minimum wear and tear of an engine may be secured by proper attention and periodic inspections, and that when a carefully planned routine schedule is followed, these maintenance problems are more easily handled.

The following maintenance schedule for Diesel engines, which was presented in this year's report of the Prime Movers Committee of the National Electric Light Association, is a guide for periodic inspections and the maximum time limit for each. Inspections may be made more often if desired and the actual repair or other adjustments must be made as required regardless of a conflict with the routine schedule. The inspections as summarized and formulated are meant to serve more as suggestions rather than instructions.

OUTLINE OF MAINTENANCE SCHEDULE FOR DIESEL ENGINES

Inspections	Time Interval
1. Piston and cylinder	Every 6 mo.
2. Air compressor	Every 3 mo.
3. Fuel pump	Every 6 mo.
4. Governor	Every 6 mo.
5. Cam shaft and valve lever arms	Every 12 mo.
6. Main bearing cap adjustments	Every 6 mo.
7. Exhaust valves	Every 2 mo.
8. Admission valves	Every 4 mo.
9. Fuel valves	Every 2 mo.
10. Starting valve	Every 12 mo.
11. Crankpin boxes	Every month
12. Air compressor crankpin box	Every month
13. Fuel valve setting	Every week
14. Lubricating oil system	Every 2 weeks
15. Relief valve	Every 6 mo.
16. Force feed lubricator	Every 6 mo.
17. Injection air pipes	Every 12 mo.
18. Air compressor cooling coils	Every 12 mo.
19. Main cylinder jacket	Every 12 mo.
20. Intake pipe to air compressor	Every 6 mo.
21. Fuel oil tanks and house pumps	Every 6 mo.
22. Cooling water system	Every 6 mo.
23. Air bottles	Every 12 mo.
24. Exhaust manifold and piping	Every 12 mo.
25. Engine frame, bed plate, etc	Every 6 mo.
26. Weekly inspection and report	Every week
27. Annual inspection	Yearly

It is expected that the major inspections can be planned and organized so that the least possible operating time will be lost.

When parts are removed for inspections, they should be marked plainly to facilitate re-assembling. As soon as a spare part is put into use, a new one should be ordered to replace it and the old part properly disposed of.

Whenever inspection work has been done on an engine, care should be taken to bar the engine over by hand to make sure that everything clears properly and that the engine is ready for operation before starting.

Maintenance report blanks are furnished with this schedule and care should be taken by the chief engineer to note and report, in detail, conditions as found while making the inspections outlined.

Change in Furnace Setting

IN ANSWER to F. G. F.'s question on page 796 of the Aug. 1 issue of Power Plant Engineering, if he will raise his boiler 24 in., leave the bridge wall as it is and remove 6 or 8 in. more of the lower baffles, he will have a good setting. Inclined grates would be of no use in wood burning. L. B. S.

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CONTENTS

Uneeda Bakers Start Up New Buffalo Plant. Illustrated.....857
Some Notes on Centrifugal Pumps. By C. C. Brown. Illustrated 867
Insulation of Cold Surfaces to Prevent Sweating. By L. L. Barrett. Illustrated.....870
Practical Construction of Alinement Charts—II. By R. T. Livingston. Illustrated.....873
Automatic Control for Oil Fired Boilers. By S. B. Derby. Illustrated 875
Failure of Steam Gage Pipe Causes Some Excitement. By H. J. Cheetham.....876
Standards for Tongue and Groove Joints. Illustrated.....877
Leaking Steam Line Repaired by Welding. Illustrated.....879
Depreciation of Equipment. By James Gibbons.....879
Oil Engine Operation Studied by Special Cards. By H. F. Birnie. Illustrated.....880
Remotorizing Plant Raises Power Factor. By O. H. Henschel. Illustrated 883
Pointers on the Maintenance of Commutators By A. R. Knapp. Illustrated.....888
Huge Portable Riveter Used by Niagara Falls Power Co. Illustrated 890
Mechanical Balance of Rotating Machinery. By Philip G. Bernholz. Illustrated.....891
Low Water? Examine Feed Line Valves. By H. A. Jahnke. Illustrated 892
A Roadman's Experience with a Mexican Revolution. By Geo. H. Wallace. Illustrated.....893
Classification of Fuel Oils.....895
Letters Direct from the Plant: Constant Excess Pressure Governor. Faulty Governor Setting. Gas Engine Valve Alterations Increases Fuel Economy. Wood Serves in Emergency as a Heavy Puller. Vacuum Breaker on Dashpot Aids in Closing Down. Adjusting Cylinder Clearance. Equivalent Pipe Sizes. Emergency Gaskets. Stopping Coal Conveyor Automatically. Burning Tires. Illustrated 896
Problems in Operating Power Equipment: Will Two Feed Water Systems Be Necessary? Scale in Feed Water Line. Use Exhaust Steam. Maintenance Schedules for Oil Engine Plants. Change in Furnace Setting.....900
Editorial: Distilled Make-Up Is Material Aid for High Boiler Ratings. Planning Your Own Advancement. At the Threshold of a New School Year. Development of Flange Standards for High Pressure Steam. Know Your Oil Engine 902
Vacuum Evaporators Used at Saxton Plant. Illustrated..904
N. A. S. E. to Meet at Buffalo.....906
Double Port Air Register Increases Oil Burning Capacity. Illustrated 906
Air Mixing Feature Part of Pulverized Coal Burner. Illustrated 907
Single Pass Condenser in Plant of Pennsylvania Edison Co. Illustrated 907
Steam Trap Made of Boiler Plate and Steel Pipe. Illustrated 908
Potential Transformer Fuses for Indoor Use. Illustrated.908
World's Largest Fuel Oil Truck Delivers Oil to Power Plants. Illustrated 908
Weymouth Station Site Chosen from Airplane Photographs. Illustrated 909
Water Power Development for Pennsylvania.....909
$30,000,000 Development Planned for White River.....909
News Notes 910
Catalog Notes 910

Distilled Make-up is Material Aid for High Boiler Ratings

Evaporators as applied to the distillation of feed water make-up are coming into more common use. This is due in large measure to the increased use of motor-driven auxiliaries with resulting lower percentage of make-up water required. The problem of evaporator application depends principally upon whether or not conditions of operation and maintenance will show that a saving can be made, due to evaporator use.

Where the make-up water does not exceed 15 per cent of the total water evaporated by the boilers, the use of evaporators has, in many cases, been a direct advantage when conditions require operation at high overloads for sustained periods of time. Condenser leakage must be kept at a minimum, otherwise the advantages of distilled make-up water may be entirely off-set. Condensate free from foreign matter and distilled make-up water mean: Less danger from tube failures; clean steaming surfaces with resultant greater rapidity of steaming and greater economy of fuel; a saving due to blow-down losses and a saving in labor hours because shutdowns for cleaning boilers will become less frequent. This last saving also means that the boilers will become more productive, in that they can remain in continuous service for a comparatively greater period of time.

These advantages have attracted the engineer to a study of the adaptability of evaporators to his specific problem. He has found it an economic problem which involves first a decision that conditions are favorable to the use of evaporators. It then follows that an analysis must be made to determine whether high or low-pressure evaporators should be used and whether the "high" or "low" heat level system is best adapted to the general scheme of heat balance which it is desired to secure.

Practical evaporator applications covering a wide range of conditions have now been made and there is probably no reason why the problem of the use of distilled make-up water cannot be successfully approached both from the theoretical and the practical standpoints.

Planning Your Own Advancement

At some period in his life, every man feels that he ought to go into some business of his own—work for himself and not be dependent on a salary or wages. This ambition, worthy as it may be, can never, in the majority of cases, be realized.

In the first place, economic stability demands that there be a certain number of directed workers; in the second place, a majority of workers are not fitted either by training or temperament for the successful operation of their own business establishments. The high mortality rate among business enterprises is indicative of this latter point.

How, then, may a man who has a definite training along certain lines achieve the greatest success, both mentally and financially?

While advice on this general subject is plentiful, there is one angle of it which has not received as much comment as it deserves. The point is this: look upon your experience and training as a business. Take an occasional inventory of yourself just as the merchant

does. Is your mental capital greater than it was a year ago?

If you were to capitalize your mental assets, at what figure could you conscientiously value them? If your mental capacity is worth $30,000, then allowing 10 per cent as a fair charge for interest, profit and depreciation, your annual salary ought to be $3000 a year.

Many engineers whose valuation may be far in excess of that figure have incomes less than the $3000 figure. The reason for this state of affairs is obvious; their sales department has not kept pace with production.

In operating a successful enterprise, the business man looks ahead and analyzes every move with a view to its effect on the profit and loss account. Engineers ought to apply the same method profitably to their own advancement.

At the Threshold of a New School Year

Within the next few weeks, some 60,000 young men will enroll in the technical schools of the country. Of these, some 36,000 will register for the work of the first and second years, while 14,000 will enter their third year and 10,000 will return for the fourth year. Mechanical engineering will claim about 14,000 of these, electrical engineering 13,000, civil engineering 13,000 and chemical engineering 7000. The others will be distributed throughout the wide range of the technical professions. Although these figures are approximate, they are based on the registration figures for 1922 and 1923 compiled by the United States Bureau of Education.

In potential influence, this army of young men represents a power for scientific advancement which cannot be measured. Who shall say but from their ranks may spring one whose contributions to the advancement of mankind will more than offset the cost of educating the entire group. Many, it is true, will fall by the wayside; some will enter other lines of endeavor; but considered in its entirety, the group will work its way to the ultimate goal.

It is not alone from the standpoint of learning the fundamentals of a particular profession that technical education is important. It is the development of logical, analytical thinking, which is the ideal of the engineering school.

Students, and frequently instructors, are prone to place too much importance on shop work, on drafting and other of the mechanical details. It is not the purpose of the technical school to turn out finished tradesmen; that can be accomplished more rapidly, and with less expense, on the job.

While the development of logical thinking as applied to engineering problems is important, that in itself is not sufficient. Unless the scheme of education trains the student to apply the same method of thought to general business problems, including his own advancement, the school itself is at fault.

An engineer who cannot write a business letter, who is not versed in economics, who does not understand the problems of labor, who, in short, has missed getting the modern executive viewpoint, has not fitted himself for making the most of his engineering training.

Development of Flange Standards for High Pressure Steam

Substantial design suitable for the work to be performed and a full consideration of what the design will mean in ultimate cost are two factors which stand out pre-eminently in the development and adoption of standards.

Absence of adopted standards for flanges and bolting for use with high pressure steam has caused considerable confusion. Many engineers have met conditions by the establishment of individual standards which vary greatly in the details of design and which lack uniformity as to range of standards. This condition has resulted in high cost of production.

In this issue and the one preceding are offered suggested standards by Olaf E. Oleson for flanges and bolting for steam pressures of 400 lb. and above. The design suggested takes into consideration weight of metal, placing of fillets, gasket pressure, temperature range throughout the fitting, and the convenience of erection. Attention is also called to the fact that but two standards are recommended for pressures ranging from 400 to 1200 lb. inclusive. This insures a simplicity which will reflect favorably in the purchase price and stores cost to the user.

Know Your Oil Engine

Indicator diagrams are so essential to a proper understanding of oil engine characteristics that it is surprising how many operators know little or nothing about the taking of such cards or to interpret them afterward. Steam men have made this work more of a hobby and discussions have waxed warm over the points brought out by particular cards.

In the July 1 issue, H. F. Birnie discussed the fundamental points to consider in the taking of Diesel engine cards and how they may afterward be analyzed. In the present issue Mr. Birnie continues the general subject by taking up the special type of cards known as the offset and weak spring diagrams.

Normal indicator diagrams have their value in certain studies, particularly when the indicated horsepower is involved; but they do not permit a detailed examination of those portions of the card where the pencil motion is slight as compared with the importance of the changes represented. Mr. Birnie shows how it is possible to get around this difficulty through the use of these special diagrams. He then takes a number of typical cases to illustrate the method to follow in analyzing such cards in terms of engine performance.

While it may be possible to operate an oil engine without taking indicator diagrams it is hardly probable that the operator can keep the engine in the most efficient condition without some knowledge of internal characteristics. The only means for obtaining this information is through the use of the indicator and operators of oil engines may well profit by the experience of the steam engineers in placing a proper value on the use of this instrument.

Know your oil engine is a slogan which might well be applied by engineers operating such equipment and an essential part of that slogan would be a knowledge of what is going on during the cycle of operations.

Vacuum Evaporators Used at Saxton Plant

DOUBLE EFFECT FILM EVAPORATORS OPERATE ON 0 LB. GAGE STEAM WITH 24 IN. VACUUM IN EVAPORATOR CONDENSER

EVAPORATORS in the new plant of the Pennsylvania Central Power and Light Co. at Saxton, Pa., are of the vacuum, film type and the vapor piping is so arranged that it is possible to secure (1) reversing through a small trap. The air extracted from the system is discharged from the after heater, to the atmosphere—thus all exhaust and drain piping is eliminated and the unit is perfectly self-contained.

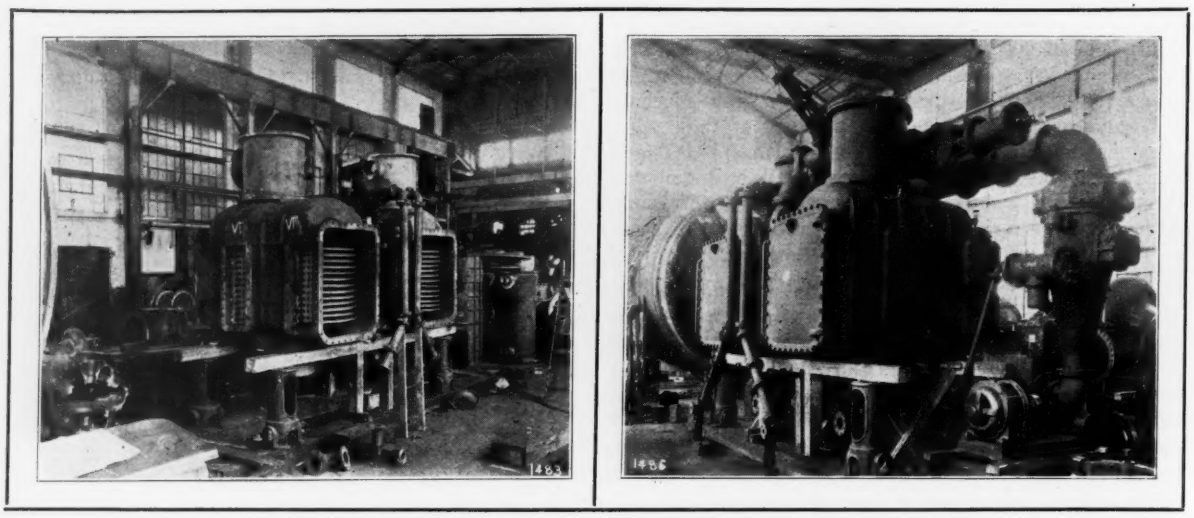


FIG. 1. SHOP SET-UP OF THE EVAPORATORS USED AT THE SAXTON PLANT

On the left is a view of the evaporators under construction showing the two drain headers from the coil banks. Air is removed from the coils by a vent pipe and control valve located at the top of each drain header, as will be noted from the view at the right. Valves in the vapor lines are operated hydraulically. It is possible to reverse the double effect operation.

double effect, (2) either evaporator as a single effect and (3) both evaporators as single effects.

Day & Zimmerman, Inc., are the designers of the station and this equipment was selected by them, working with the engineers of the Wheeler Condenser & Engineering Co. It consists of a Wheeler film evaporator, double effect unit, of the Ellipti-coil type. A low level jet condenser serves to condense the distilled vapors from the evaporator; the main condensate, condensed vapors, and evaporator coil drains all being combined and discharged by the condenser removal pump.

Normally the unit will operate with exhaust steam, at 0 lb. gage pressure, and carry a vacuum in the evaporator condenser of approximately 24 in.

Air is removed from the system by a single-stage steam jet air pump, mounted on the evaporator condenser. The small amount of steam consumed by the jet, approximately 140 lb. per hr., is condensed by an after heater and drained into the evaporator condenser

Figure 2, showing the heat chart, indicates the approximate station heat balance, and the arrangement of the evaporator and evaporator condenser. Main condenser condensate serves to condense the exhaust steam from the steam jet air pump, and then passes into the evaporator condenser, regaining all the heat supplied the evaporator. The combined condensate make up then is pumped to the station heater, of the jet type, where the remainder of the auxiliary exhaust steam is condensed and the boiler feed pump pumps this water to the economizers, at approximately 190 deg. F.

Since economizers are installed in this station, it is imperative that no excessive heating of the feed water be allowed, in order to get maximum efficiency from the economizer unit. Consequently the ideal arrangement, in this case, is the use of exhaust steam for the evaporator supply.

Thus the evaporator unit does not affect the overall station heat balance at all; steam which would otherwise

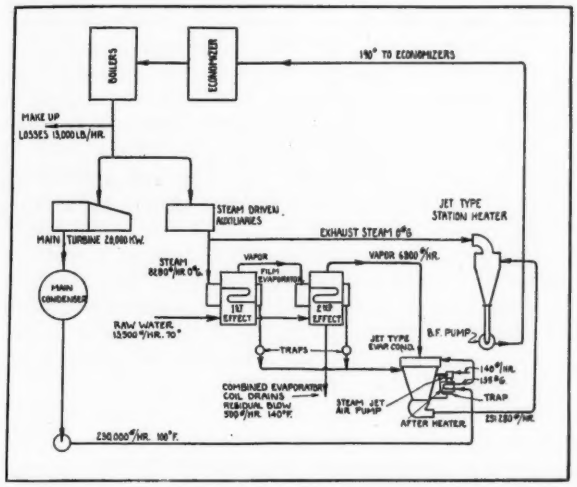


FIG. 2. APPROXIMATE HEAT CHART OF THE EVAPORATOR SYSTEM

go to the station heater simply produces the distilled water in the evaporator, and the heat is absorbed by the main condensate in the evaporator condenser. With the evaporator shut down, all the exhaust steam passes direct to the station heater producing the same final feed water temperature to the economizers, as is the case with the distilled water unit in operation.

The film evaporator is of the Ellipti-coil design, with the heating surface made up of a series of coils. In the double effect unit, the two evaporators are identical in design and construction, as well as to sizes of inlet and outlet connections.

Each evaporator consists of a rectangular cast iron shell, containing two banks of coils. Each bank has a steam header on one side of the shell, and a drain header on the opposite side. The ellipti-coils are arranged in units of three coils each. These three coils, all in one horizontal plane, have a manifold flange at the steam header and the drain header, each held in place by tap screws, which are inserted from the coil headers. Thus each Ellipti-coil unit can be readily removed from the evaporator by removing the tap screws. One bank of coils is accessible from one door, the other bank from the door at the opposite end of the evaporator.

In this type of evaporator the coils are not submerged in the raw water being evaporated. The water level is maintained just below the bottom coil unit by a float controlled water inlet valve.

Heat transfer from the coils to the raw water is accomplished by pumping the water to the distributing plate, where upon it sprays down over the tubes to the well. In this way, the water is constantly recirculated. The distilled vapors separate from the films without ebullition and pass smoothly to the vapor dome.

Coil headers are drained to the evaporator condenser, the pressure drop serving to insure a positive flow. Condensate traps of the Corliss valve type control the flow from the drain headers. Shut-off valves are provided for cutting out one evaporator. The drained condensate partially flashes into vapor in the evaporator condenser, being thoroughly deaerated in the process.

In designing the jet type condenser installed with this unit, special baffles and an extra large shell are used to insure a minimum of air passing out through the removal pump with the condensate. With a vapor temperature of 140 deg. F., the temperature of the outgoing condensate with this design is 138 deg. to 139 deg. F. or the terminal difference is 1 deg to 2 deg. F. Thus if proper precautions have been taken to prevent the mechanical entraining of air in the tail water discharge, condensate is well deaerated. A steam jet air pump of

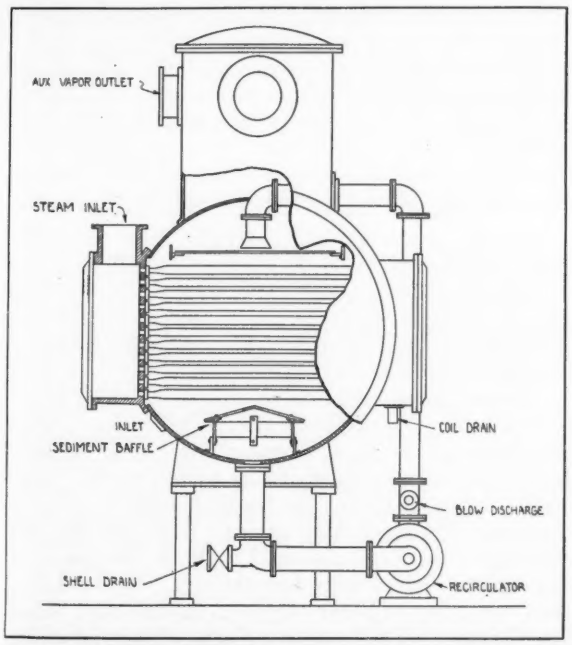


FIG. 3. WATER TO BE EVAPORATED IS RECIRCULATED TO THE DISTRIBUTING PLATE WHERE IT SHOWERS DOWN OVER EVAPORATOR TUBES

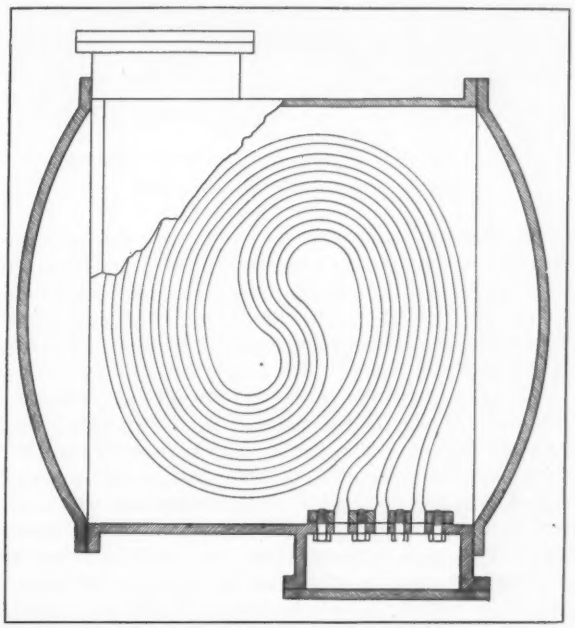


FIG. 4. COILS ARE ARRANGED SO THAT WEAVING MAY TAKE PLACE WITH PRESSURE AND TEMPERATURE CHANGES

an oversize capacity removes the air. This is shown, mounted on the side of the condenser in Fig. 1 with the after heater and rain trap.

Continual discharge of the concentrated sludge, or raw water concentrate, is maintained by a small centrifugal blow pump, driven through one of the evaporator recirculators. The power requirements of this pump are practically negligible.

Double Port Air Register Increases Oil Burning Capacity

DURING the past year, the Peabody Engineering Corporation, of New York City, has developed a new design of air register which makes it possible to burn considerably more oil with the same draft. This register, known as the Peabody double port register, was designed for oil burning installations where the draft available is so low that satisfactory ratings can not be obtained from the boiler plant.

In a great many cases, it is impractical to use forced draft or to increase the stack height in order to obtain sufficient draft for the ordinary air register. Such

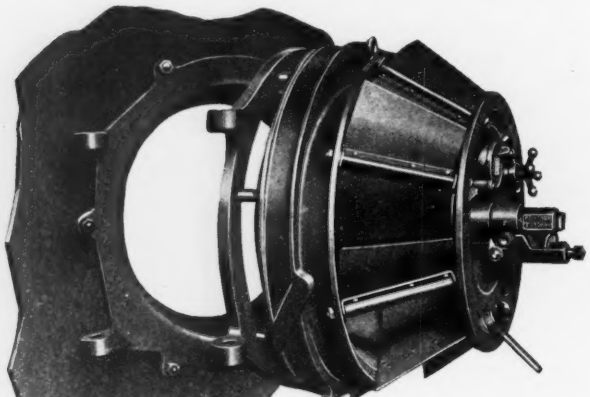


FIG. 1. AMOUNT OF OPENING OF THE SECONDARY AIR PORT IS CONTROLLED BY A RING WHICH IS OPERATED BY A CRANK MOTION AND LEVER

changes would necessarily be quite expensive, as compared with the cost of a natural draft installation applied to the existing plant. It is also sometimes impossible to increase the number of burners per boiler due to restrictions in furnace volume and in the size of the furnace front wall.

In this register the outer part, forming the frustrum of a cone, is exactly the same as in the standard Peabody air register. The double port register is made up simply by adding a curved deflector casting and a conical shaped furnace casting or mixing chamber, which are held at the proper distance from each other by spacer bolts. The space between these two castings forms an auxiliary air opening, of substantial area, on the furnace end of the register, hence the name "double port" register. This auxiliary air opening is adjusted by means of a cast-iron closing ring actuated by a crank shaft at the bottom of the register. Angular slots in the ring are guided by rollers on the outer rim of the deflector so that as the ring closes, it also revolves, thereby eliminating any tendency to stick or bind. The operating shaft is provided with a handle located at the bottom of the register front plate, and also with a locking handle so that the ring may be set at any desired position and kept there. The regular curved doors of the front or fireroom end of the register are adjusted and set by means of a hand-wheel and locknut on the upper part of the register front plate.

For land installations, the fixed type of register is furnished, as it is generally preferable to enter the furnace through an access door built in the side wall of the furnace. The hinged design, Figs. 1 and 2, is furnished for marine jobs or Scotch boiler installations where ready access to the furnace is desirable.

It has been found that it is possible to burn approximately one and one-half times as much oil per burner per hour with the double port register as with the ordinary type of register with the same draft. The register requires only two adjustments, and with a little practice, the operating force is able to maintain the highest efficiency.

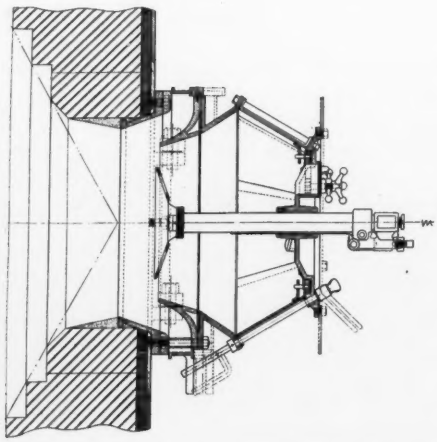


FIG. 2. FIFTY PER CENT MORE OIL MAY BE BURNED WITH THE SAME DRAFT THROUGH USE OF DOUBLE-PORT REGISTER

N. A. S. E. to Meet at Buffalo

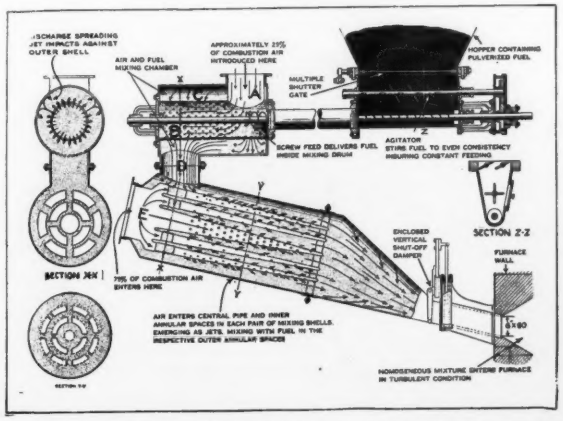
DURING THE week of September 10 the National Association of Stationary Engineers will hold its annual convention at Buffalo, N. Y. Sessions of the association and the exhibit of power plant equipment, which is sponsored by the National Exhibitors Association, will be held at the Broadway Auditorium.

Among the subjects to be discussed which are of general interest to engineers are the following: "Plant Maintenance," by T. A. Fleming, director of conservation of the National Board of Fire Underwriters; "Automatic Control of Fires," by Ira G. Hoagland, secretary Automatic Sprinkler Association; "Underwriters Laboratories Service to Engineers," by G. B. Muldaur, general agent, Underwriters Laboratories. W. L. Abbott, chief operating engineer of the Commonwealth Edison Co., Chicago, will talk on some phases of the coal storage investigation now under way by the committee appointed Federated Engineering Societies. Mr. Abbott is chairman of the committee.

Air Mixing Feature Part of Pulverized Coal Burner

ONE of the problems in the successful use of pulverized coal as a fuel is that of introducing the combustion air in such a way that all of the minute particles of coal will come in contact with sufficient oxygen to complete combustion. Several methods have been worked out to meet this problem, among which is a type of burner called the "Multi-Mix," developed by the Ground Coal Engineering Co.

In the accompanying illustration is shown the method of diffusing coal dust through the combustion air in the "Multi-Mix" burner. The coal dust hopper is connected above the casing in the upper right hand corner


DISCHARGE SPREADING JET IMPACTS AGAINST OUTER SHELL
AIR AND FUEL MIXING CHAMBER
APPROXIMATELY 25% OF COMBUSTION AIR INTRODUCED HERE
HOPPER CONTAINING PULVERIZED FUEL
MULTIPLE SHUTTER GATE
SCREW FEED DELIVERS FUEL INSIDE MIXING DRUM
AGITATOR STIRS FUEL TO EVEN CONSISTENCY INSURING CONSTANT FEEDING
SECTION Z-Z
ENCLOSED VERTICAL SHUT-OFF DAMPER
FURNACE WALL
SECTION X-X
75% OF COMBUSTION AIR ENTERS HERE
AIR ENTERS CENTRAL PIPE AND INNER ANNULAR SPACES IN EACH PAIR OF MIXING SHELLS. EMERGING AS JETS, MIXING WITH FUEL IN THE RESPECTIVE OUTER ANNULAR SPACES
4'x60
HOMOGENEOUS MIXTURE ENTERS FURNACE IN TURBULENT CONDITION

AIR REQUIRED FOR COMBUSTION IS INTRODUCED BEFORE PULVERIZED COAL ENTERS FURNACE

of the illustration. The screw is rotated by suitable driver and delivers the coal into the space B inside the perforated screen of the mixing drum.

Only a relatively small portion of the total combustion air is introduced at A and the blast goes directly into the coal as it is whipped off by the screw, picking it up and carrying it outward through the perforations in the mixing screen. Each small stream of air and coal going through its hole in this screen is subjected to a kneading or wiredrawing effect, tending to diffuse the particles uniformly in the air.

On emerging, the jet impinges against the outer shell and is broken up, cross firing into other jets, thus setting up a mass of eddying currents in the annular space C around the screen. All of these currents are then drawn together and delivered through the connection D to the main mixing part of the burner.

When the partially mixed material comes in at D it is distributed through the communicating tubes to the alternate annular strata while the remainder of the combustion air is admitted at E into central perforated pipe and to each of the alternate air strata, the outer shell in each case being perforated so that the air passes through the perforations as jets and thus completes the even mixing of the material with all of the air theoretically required for combustion. With this mixing effect, no excess air need be used.

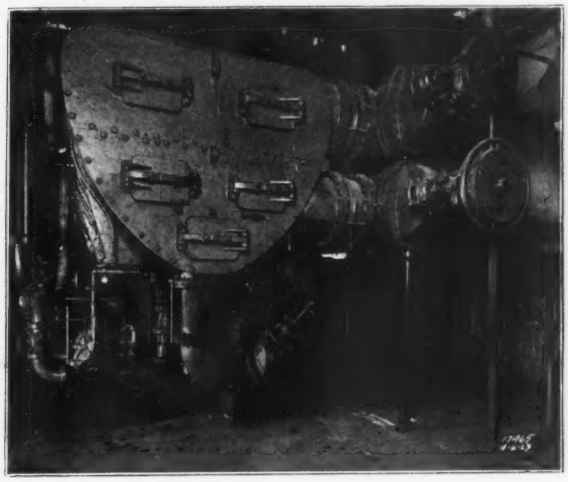
It should be noted that air only comes from the center through the holes in mixing shells, the coal being at all times outside and surrounding the shells; and free to travel outward to the nozzle through a constantly increasing area of passage, so that clogging is impossible.

As the outer annular strata of each mixing pair is of larger volume than its corresponding inner air strata, the relatively small proportion of total air admitted at A finds itself in a space volume larger than that of the large proportion of air admitted at E. Thus the relatively small volume of coal laden air tends to expand and set up an ideal condition for the piercing air jets coming through the shell perforations to penetrate and spread as they enter the mixture, thereby bringing about an ideal mixing condition.

In January of this year, the Ground Coal Engineering Co. was acquired by the Sanford Riley Stoker Co., of Worcester, Mass., and this organization is now handling the "Multi-Mix" method of pulverized coal burning.

Single Pass Condenser in Plant of Pennsylvania Edison Co.

DURING the past winter, the first large surface condenser of the new single pass type was installed by the Ingersoll-Rand Co. in a central station plant. This unit,



TUBES IN SUCCESSIVE STAGES ARE ARRANGED ON SHORTER CENTERS

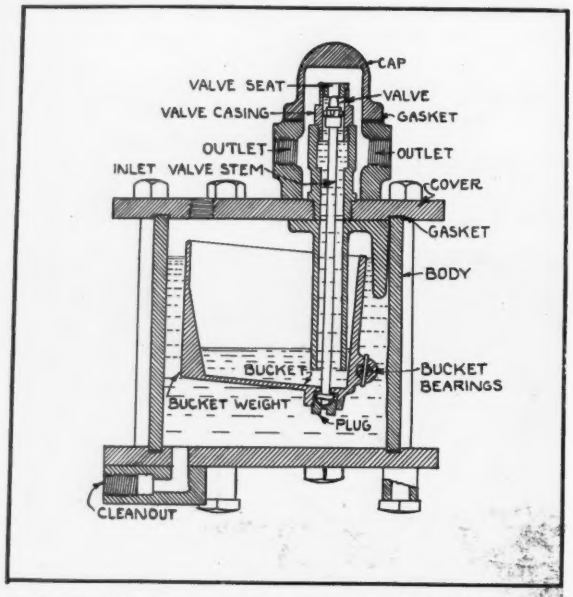
shown in the accompanying illustration is operating in the Dock St. plant of the Pennsylvania Edison Co. It is rated at 10,000 kw. The view is from the inlet water box end. Note the arrangement of condensate pumps, spring supports on one side only, motor controlled inlet valves to water box and quick opening doors on water box. The development of this type of condenser was described in the March 1, 1922, issue of Power Plant Engineering.

CALIFORNIA-OREGON POWER CO. has announced that work will begin immediately on a new plant for Klamath Falls, Ore., to cost approximately $1,500,000. Heretofore it has been necessary for the company to transmit power from Copco, Cal., to take care of the local demand and the new plant is expected to carry the entire Klamath load. Sawmills are heavy users of electric power here as there are more than 30 mills in the vicinity.

Steam Trap Made of Boiler Plate and Steel Pipe

IN ORDER to meet the modern plant conditions of high boiler pressure and superheated steam, a steam trap has been developed recently which is built of boiler plate and steel pipe. As will be noted from the accompanying illustration, the body of the trap is a piece of steel pipe while the ends are made up by plates which are held together by bolts. Gaskets are used in making pressure type joints.

It is claimed that the advantage of this type of construction over the use of steel castings lies in the fact that such castings may be porous and unable to withstand the strain of high pressure and high temperature work.



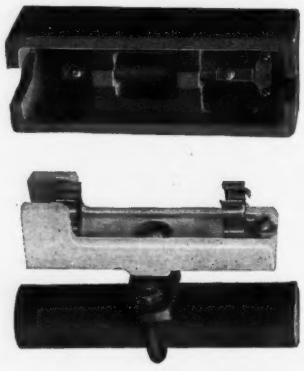
SHORT PIECE OF PIPE WITH END PLATES COMPOSE THE BODY OF TRAP

This trap is built by the C. E. Squires Co., of Cleveland, and operates on the familiar tilting bucket principle.

Potential Transformer Fuses for Indoor Use

THERE HAS recently been placed upon the market by the Westinghouse Electric & Manufacturing Co. a line of enclosed cartridge type potential transformer fuses and fuse blocks in voltages of 2500 to 25,000 for the protection of indoor potential transformers. They can also be used to protect other circuits where the current does not exceed one-half an ampere.

These fuses can be applied without preventive resistances in locations where the maximum short-circuit current does not exceed the interrupting capacity of the fuse. When a preventive resistance is used, the short circuit current will be limited, irrespective of the power back of the fuse, to a value within the interrupting capacity of the fuse. For 15,000 and 25,000-v. service the same fuse is used with a different preventive resistance to limit the current and a different base for each of the two voltages. The 2500-v. fuse base is provided with a cover of molded insulation which holds the fuse and provides for its safe handling. The higher voltage bases use corrugated post type insulators.

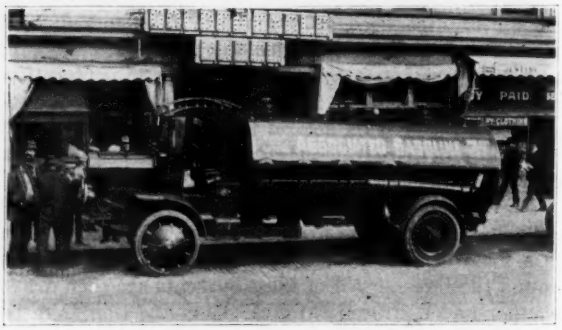


2500-V. FUSE BLOCK, OPEN TO SHOW METHOD OF MOUNTING THE FUSE IN COVER

Some of the special features of these fuses and fuse blocks are their high interrupting capacities, strong non-absorbent fiber casings, and their construction providing for the proper venting of gases.

World's Largest Fuel Oil Truck Delivers to Power Plants

FOR THE purpose of delivering fuel oil to four large plants in San Francisco so that they would not be troubled so often, the Associated Oil Co. is operating a specially designed fuel oil truck that can carry 2700 gal. at one trip, which as far as known is the largest fuel oil truck ever built. This truck delivers fuel to the power plants of the largest hotel in San Francisco, the largest laundry and two power plants of the Pacific Gas & Electric Co. This truck is not only remarkable in capacity, but also in the originality of its design. The standard McDonals front wheel drive assembly has been used with the addition of a steering control by means of hydraulic equipment. The truck has a 50-hp. motor, which is assembled with the transmission and differential on an unusually heavy sub-frame. The sub-frame is attached to the main frame by four quarter-elliptical springs.



FUEL OIL TRUCK HAS A CAPACITY OF 2700 GAL.

Weymouth Station Site Chosen from Airplane Photographs

IN MAKING a selection of a site for its new power station, the Edison Electric Illuminating Co., of Boston, resorted to use of airplane photographs. Air photographs were taken of many prospective sites within a wide radius of Greater Boston. Following a study of these views, the Weymouth plot was selected as the best.

The site of the new plant is the tip of a peninsula extending into the Fore River. The Fore River plant of the Bethlehem Shipbuilding Corporation is directly across the deep channel separating this peninsula from the Quincy (Mass.) shore. The peninsula is 2 mi. from the open harbor at tidewater. The channel leading to it allows unobstructed approach for coal barges to the plant's storage area for coal, which will be located along the channel. This area will have a capacity of 300,000 T.

WEYMOUTH STATION TO BE LOCATED ON POINT OF LAND MARKED WITH A CROSS

In determining the most advantageous layout for the various units of the plant, airplanes were employed further to good use. For example, aerial views revealed that it was not only advisable but necessary to change the location of the main transmission line to lead from the station as plotted on the original plans.

According to plans, the new station eventually will become the main link in the electrical system operated by the company in supplying power and light to Boston and 41 other cities and towns in a territory totaling 616 sq. mi. The station will be hooked up with the present central station of the company 10 mi. away in Boston. This station, known as the L Street Station, has a generating capacity of 248,000 hp. and has reached its limit.

$30,000,000 Development Planned for White River

PLANS are under way by the North American Co. for a $30,000,000 hydroelectric development project on the White River, in the heart of the Ozark Mountains in Arkansas and Mississippi. The plans call for the development of 160,000 hp. and embrace a high tension transmission line of some 240 mi. to St. Louis.

Hugh L. Cooper & Co., it is stated, has secured an option on all the rights and holdings of the Dixie Power Company for the development of water power on the White River in Arkansas and Missouri, and under contract for the North American Company has commenced a complete study of the surveys, water supply data, overflow damages, etc., and will make the necessary diamond drillings of the dam site. These studies and explorations should be completed by fall, and the beginning of the actual construction work early in 1924 is a possibility.

According to officials of the North American Co., the White River Development has been under consideration for more than 10 yr. President Taft in 1912 refused the Dixie Power Co. a perpetual franchise. Then followed several years of struggle for Federal water power legislation, and in 1920 the Federal water power act became a law. The White River project was granted Permit No. 1.

An extensive survey of the site has been made by the Dixie Power Co., involving a study of the whole White River terrain from Cotter to Forsyth. The watershed above Cotter is 6200 sq. mi., with an average annual rainfall of about 45 in.

It is proposed to construct a special arch-type concrete dam 225 ft. high, its base being about 1000 ft. in length and 1800 ft. at the crest, establishing an artificial lake nearly 100 mi. long and covering over 50,000 acres. After the power requirements in the vicinity have been taken care of the balance of the power probably will be transmitted to St. Louis, to be used by the Missouri subsidiaries of the North American Co.

Water Power Development Planned for Pennsylvania

INTERESTS associated with the Pennsylvania Power and Light Co. have bought the Pennsylvania, New York and New Jersey Power Co. and plans are now being made for the immediate development of the power site formerly owned by that company on Wallenpaupack Creek, located in the northeastern part of the state, near Hawley.

Plans contemplate the building of a hydroelectric development to have an installed generating capacity of 40,000 kw. The work in connection with this development will include a concrete dam 1150 ft. long with a maximum height of 50 ft. above the river bed. This dam is expected to create a reservoir about 12 mi. in length covering an area of 5600 acres.

Water will be conducted through a pipe line 14 ft. in diameter and about 4 mi. long, to a surge tank and thence by two 8-ft. steel penstocks, each 750 ft. long, to the power house, which is to be situated on the south shore of the Lackawaxen River.

Electric energy will be fed 50 mi. to the south into the Pennsylvania Power and Light Co.'s present system over several transmission lines, one of which is to be a 220,000-v. steel tower line.

It is estimated that the final cost of the complete development, together with the transmission line to be constructed, will be approximately $8,000,000. Construction work which will take about 2 yr. to complete, will be begun as soon as possible in order that water may be impounded in the reservoir next spring.

News Notes

HEINE BOILER CO. announces that George F. Murphy, having completed the reorganization of the New York office, has taken charge of the Philadelphia territory with headquarters in the Pennsylvania Building, Philadelphia. Harold P. Childs, formerly special representative of the executive offices of the General Electric Co., New York, has been appointed manager of the New York office with headquarters at No. 11, Broadway. J. R. Fortune formerly manager of the Detroit office, has assumed charge of the territories covered by the Pittsburgh, Cleveland and Detroit offices and will maintain offices in the Dime Bank Building, Detroit and in the Park Building, Pittsburgh, the Cleveland office having been discontinued.

JOHN HOYES MCGOWAN, president of John H. McGowan Co., pump manufacturers, Cincinnati, died at his home in that city Aug. 3, aged 93 yr. He was born in Scotland in 1830 and in 1832 his parents moved to America, finally settling in Cincinnati in 1836. In 1862, with his brother, T. J. McGowan, he formed the present company, which was incorporated in 1881. He served for 41 yr. as president of the company and retired from active participation in its affairs several years ago. He is survived by seven children.

ORDERS FOR new equipment totaling about $1,000,000 for electrifying 35 additional miles of the Paulista Railway, Brazil, have just been placed with the International General Electric Co. This extension brings the total electrified mileage up to 64 mi. starting at Jundiahy and extending north to Tatu. This is approximately half the total mileage originally planned and further extensions are expected. Included in the order, are five 62-T., 3000-v., direct current switching locomotives, a complete sub-station of 4500 kw. ultimate capacity with motor generator sets, step down transformers, switchboards, switch gear and other auxiliary equipment and overhead line material and transmission for the extension.

ON JULY 1, the control of the Sarco Co., Inc., was acquired by Clement Wells. In this connection, G. H. B. Burke, Western sales manager, has been appointed vice-president, and E. J. Ritchie has joined the board of directors.

AFTER OCTOBER 1, 1923, the Philadelphia office of Power Specialty Co., manufacturers of Foster superheaters, economizers and oil heaters, will be located in the Atlantic Building at Broad and Spruce Streets, Philadelphia.

H. S. DEMAREST, of Greene, Tweed & Co., sailed for England on Aug. 14. His plans call for an extensive trip through Great Britain and the Scandinavian countries for the purpose of visiting the Greene, Tweed agents.

PUGET SOUND POWER & LIGHT CO., a company under Stone & Webster management, has acquired control of the Olympia Light and Power Company, which supplies electric light and power and street railway service to Olympia, the state capital of Washington.

CALIFORNIA-OREGON POWER CO. has applied to the Federal Power Commission for a preliminary permit to construct four developments of the North Umpqua River, in Douglas County, Oregon, in the vicinity of Roseburg. It is estimated about 20,000 hp. will be made available.

CONTRACTS have been let for a $500,000 power plant in the construction program now being carried out by the Pittsburgh Plate Glass Co., in which $5,000,000 is being spent in making the Milwaukee, Wis., plant the largest individual paint and varnish factory in the world. The property of the Patton Paint Co., which was taken over by the Pittsburgh company, Jan. 1, 1921, is forming a large part of the new plant. The power plant will be constructed on Oregon St., near Barclay, taking in some of the ground occupied by the Milwaukee Boiler Works. It will be a 2000-kw. plant.

GEORGE B. NICHOLS, consulting engineer, 300 Madison Ave., New York City, and the Terrell Croft Engineering Co., of which Terrell Croft is directing engineer, of 6600 Delmar Boulevard, University City, St. Louis, Mo., have effected an affiliation whereby the experience and resources of each organization will be available to the clients of the other. Nichols will act as the principal for projects east of Illinois whereas the Croft organization will so act for projects west of that state. The combination will, as have its components in the past, specialize in mechanical and electrical engineering for power and industrial plants, institutions and buildings.

Catalog Notes

AMERICAN AIR-TIGHT DOORS is the title of a folder now being distributed by the Conveyors Corporation of America.

SQUIRES STEAM TRAPS are described in a booklet now being distributed by the C. E. Squires Co., of Cleveland, Ohio.

AIR COMPRESSORS and vacuum pumps are covered in bulletin No. 104 now being distributed by the Pennsylvania Pump & Compressor Co., of Easton, Pa. Details of construction are explained and typical installations illustrated.

TYPICAL EVAPORATIVE tests of Coxe travelling grate stokers, the history of the development of this stoker and a description of its features are treated in bulletin C B 2 issued by Combustion Engineering Corporation, of New York City.

LUBRICATION from Tallow to Dearborn Oil, is the title of a booklet recently issued by the Dearborn Chemical Co., of Chicago. Oil requirements for various types of service such as steam cylinders, turbines, una-flow engines, air compressors, ice machines, Diesel engines and elevators are discussed. In addition the laboratory methods used in testing oils are described.

WESTINGHOUSE ELECTRIC & MFG. CO. has issued its new 1300-page supply catalog. The material presented includes all new apparatus developed within the last 2 yr., technical data, dimension drawing, specifications and prices. All sections have been completely revised although the street lighting section shows more changes than any other section. Some 175 pages are devoted to this particular classification.